AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 42 No. 553. MAY, 1952 PRICE: 3s. 6d.

The Wickman Mobile Demonstration Unit brings skilled help right to the factory floor. Staffed by a team of specialist instructors, the Unit is fully equipped with machine tools for reservicing carbide tools, and many Wimet users have already availed themselves of this opportunity to train operators and executives in the correct methods of applying and servicing these tools. Demonstrations on the design, application and servicing of Wimet tools have been given to over 2,000 operators and, in addition, lectures, supported by sound films and film strips, have been attended by more than 1,000 operators.

The greatest productivity from carbide tooling can only be obtained when operators know the right grade to choose, the right time to withdraw tools from use for reservicing, the right wheels and methods—in fact, the right carbide technique.

This is just one of the ways Wickman's are helping Wimet users to get the best out of carbide tooling.

WICKMAN *of* COVENTRY

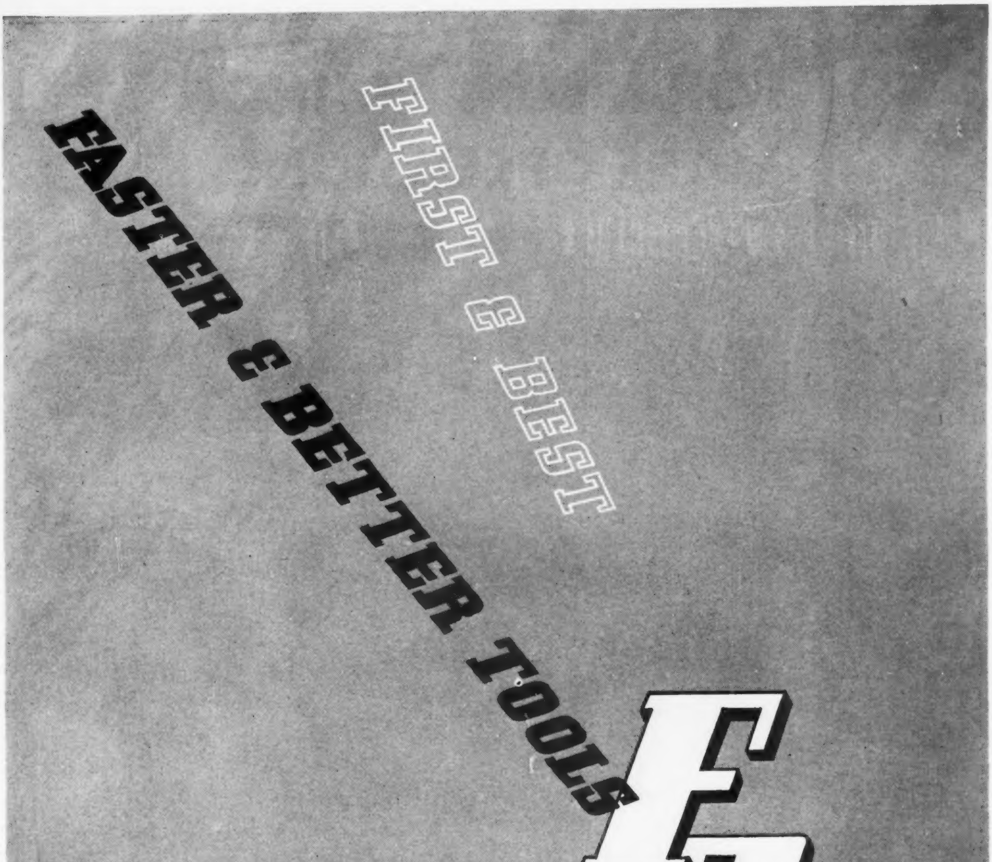
LONDON · BRISTOL · BIRMINGHAM · MANCHESTER
LEEDS · GLASGOW · NEWCASTLE · BELFAST

Wimet

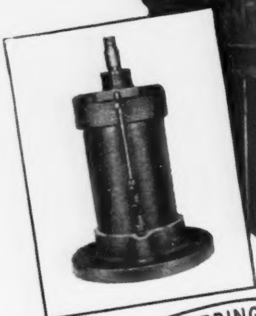
215 ST


Ryder
VERTICAL AUTOMATICS
The
No. 8 VERTICALAUTO
Maximum Swing .. $14\frac{3}{4}''$ dia.
No. of Spindles 6
No of Tooling Stations .. 5
The No. 8 Verticalauto can be arranged for double indexing or dual control A wide range of standard attachments, considerably increasing the scope of the machine, is available.
RYDER
No. 8
RYDER
Built by
THOMAS RYDER & SON, LTD.
TURNER BRIDGE WORKS, BOLTON
Wickman
Sole Selling Agents
WICKMAN of COVENTRY

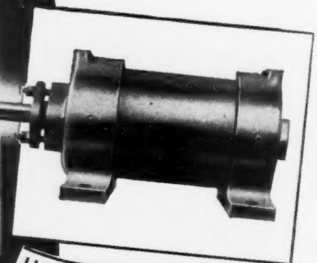
190 F18


FASTER & BETTER TOOLS
FIRST & BEST
FBT
FIRTH BROWN TOOLS
SPEEDICUT WORKS CARLISLE STREET EAST SHEFFIELD
"SPEEDICUT" CUTTING TOOLS OF ALL TYPES

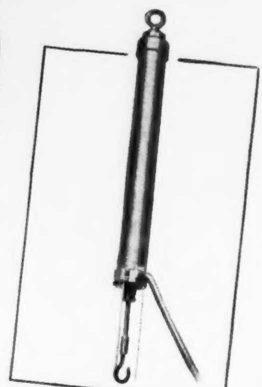
FOR MECHANISING MORE OPERATIONS



LIFTING & LOWERING



HOLDING & CLAMPING



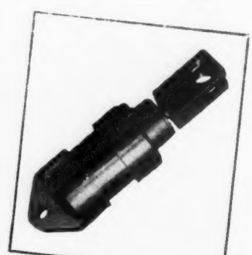
HOISTING & LOADING

On operations involving repetition movements . . . to position the work . . . to lift it . . . to move it forward . . . push it back . . . turn it round . . . and hold it steady . . . to cushion impact . . . they all add hundreds of man-hours to the total production time.

The movements cannot be **avoided**—but time taken **can** be reduced from minutes to seconds—by mechanisation with CP pneumatic hoists, pulling rams, pu hing rams, lifting cylinders and pneumatic vices. All are tailor-made to suit the specific conditions of the job.

Give your operators "flick of the finger" control over the non-productive part of their work and step up their **productive** effort by leaps and bounds. For full advice and further information . . .

CALL IN CONSOLIDATED

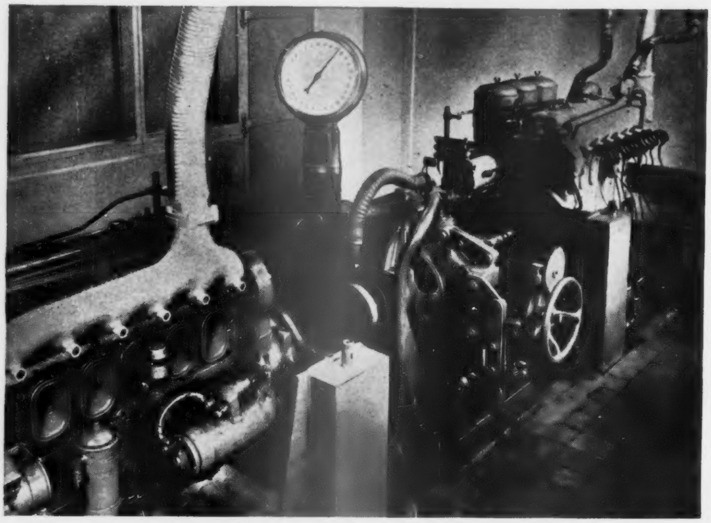


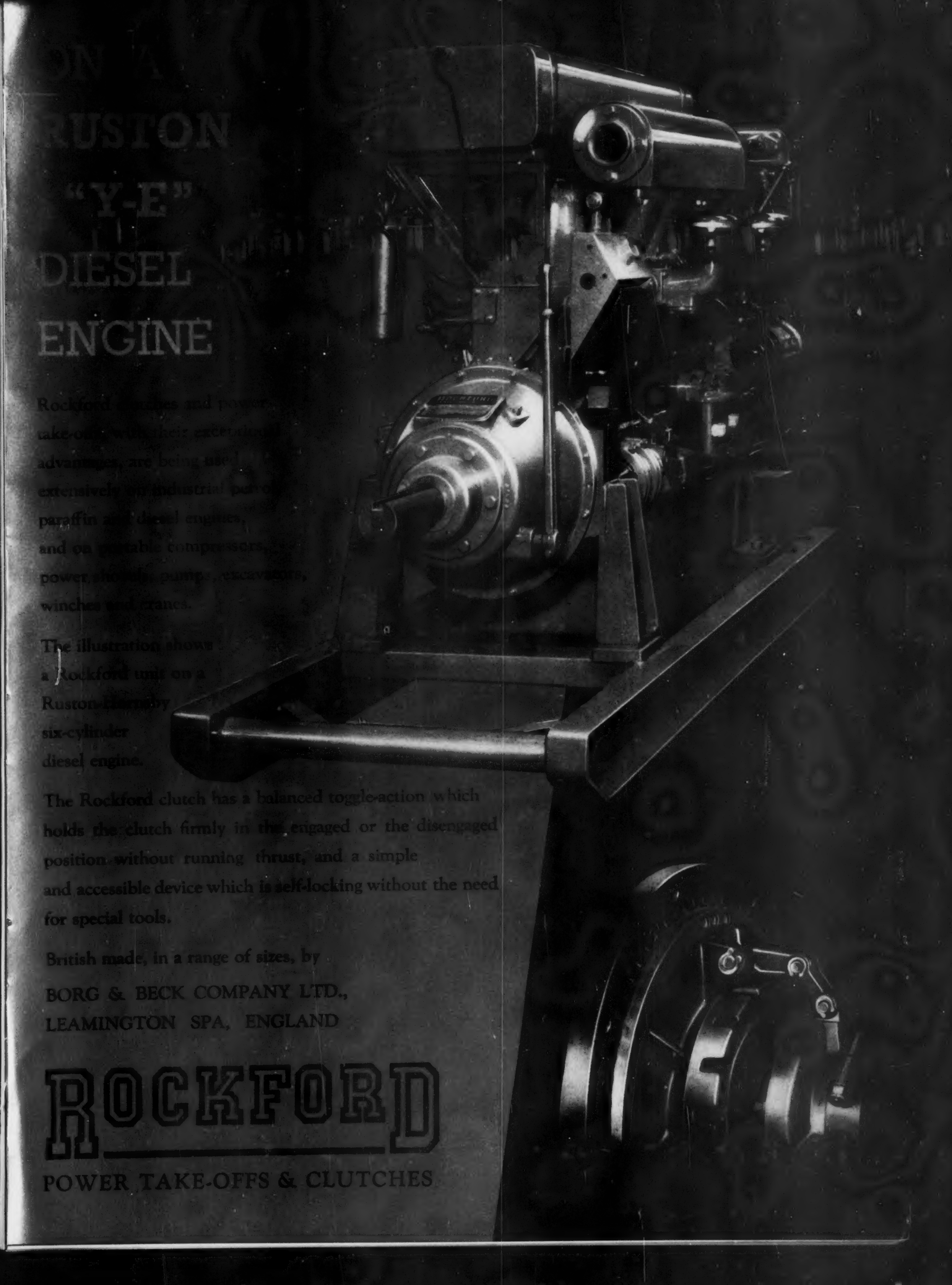
PUSHING & PULLING

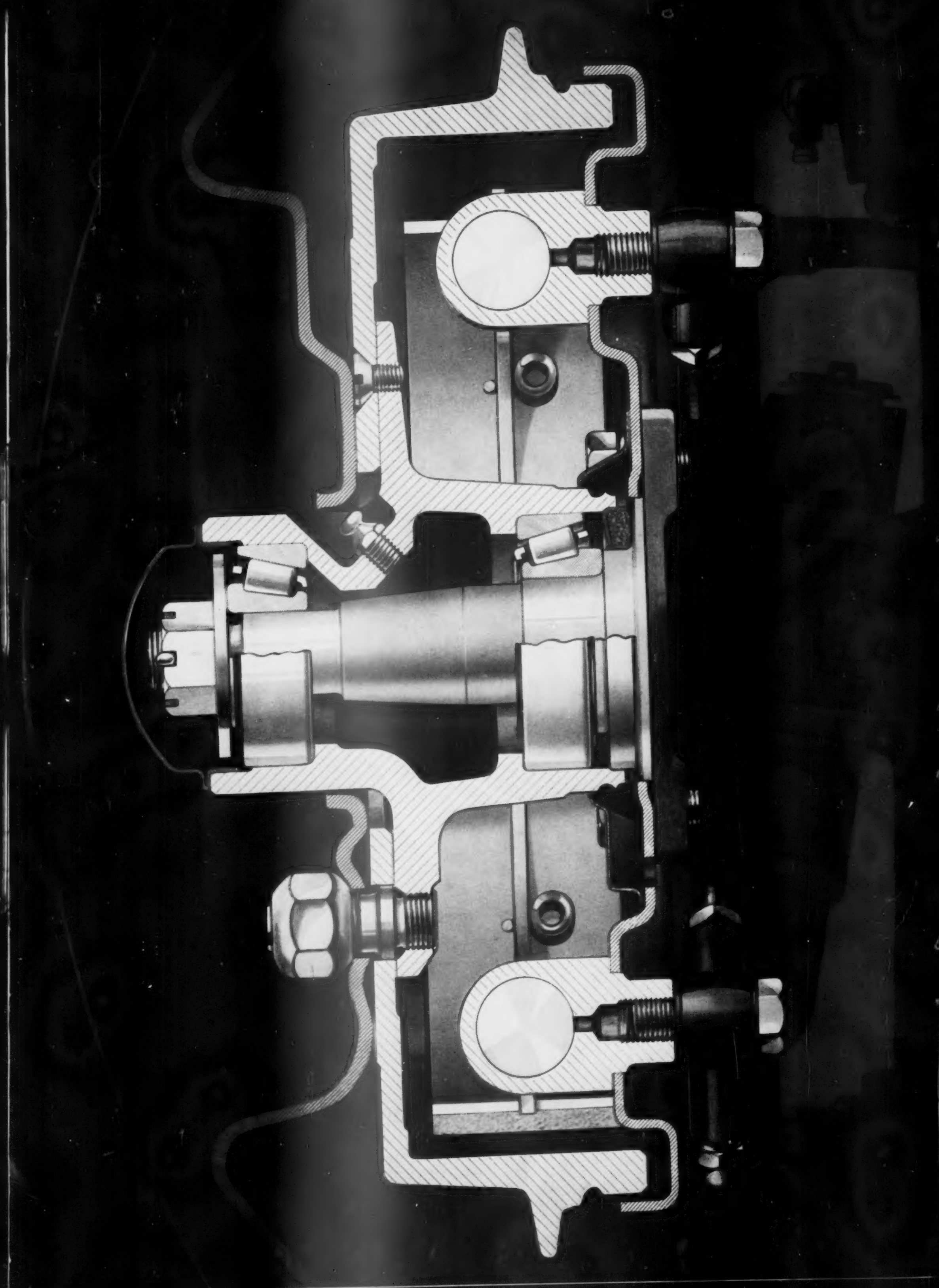
CONSOLIDATED PNEUMATIC TOOL CO. LTD.
LONDON & FRASERBURGH

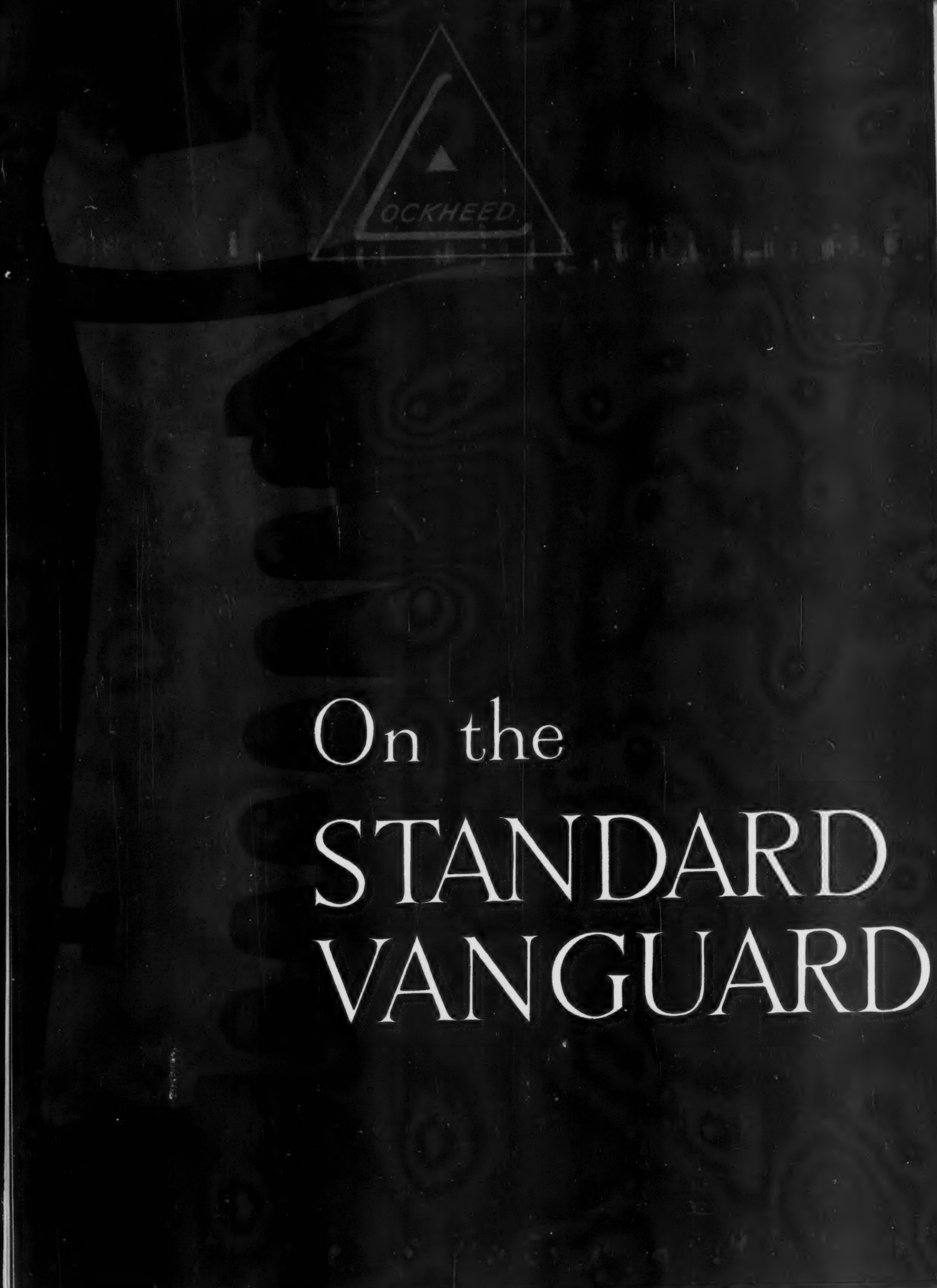
Offices at Glasgow · Newcastle · Manchester · Birmingham · Leeds · Bridgend · Belfast · Dublin · Johannesburg Bombay · Melbourne · Paris · Rotterdam · Brussels · Milan · and principal cities throughout the world.

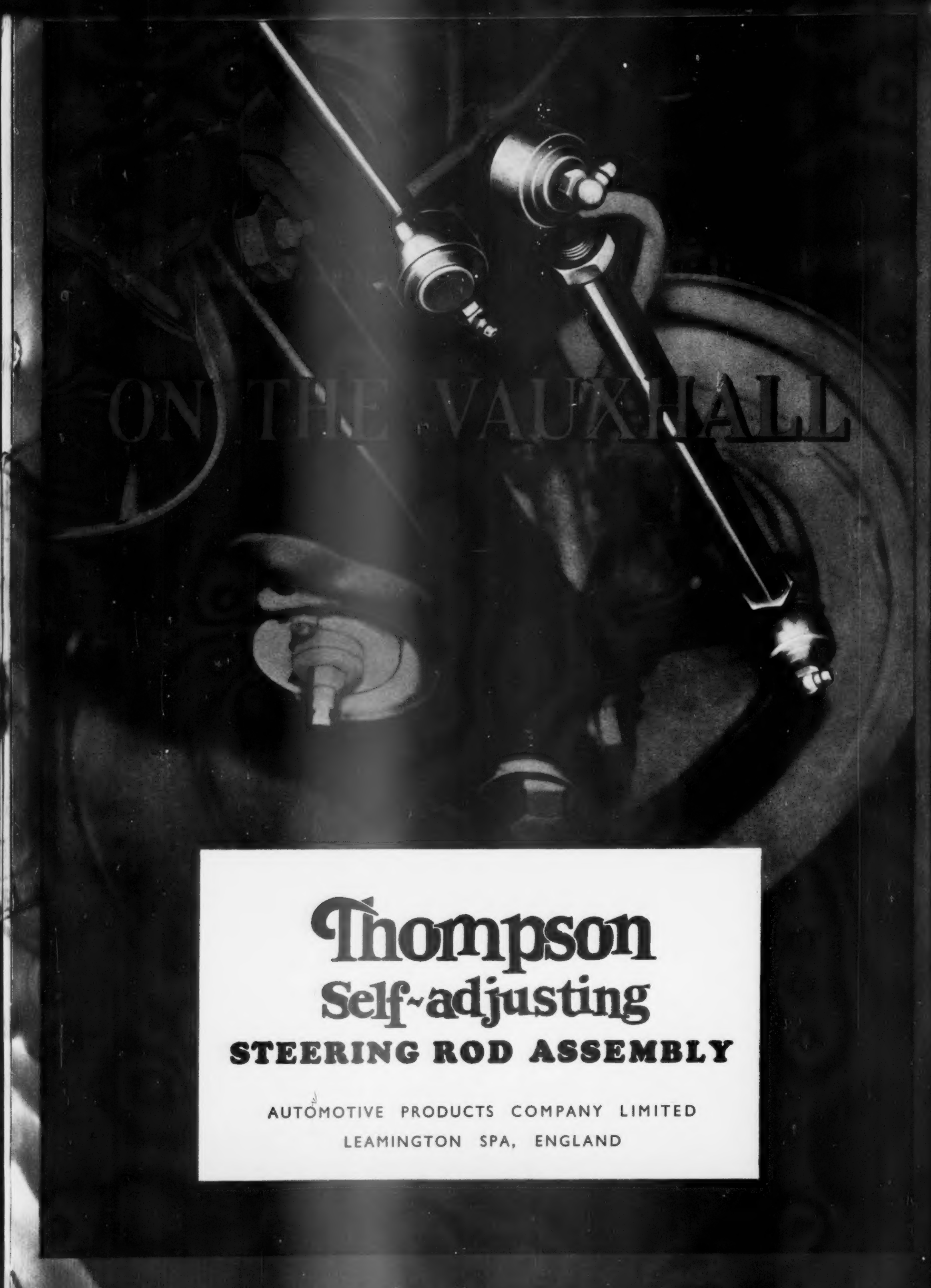
CP 112




ON A
RUSTON
"Y-E"
DIESEL
ENGINE
Rockford clutches and power take-offs, with their exceptional advantages, are being used extensively on industrial petrol, paraffin and diesel engines, and on portable compressors, power shovels, pumps, excavators, winches and cranes.
The illustration shows a Rockford unit on a Ruston-Hornsby six-cylinder diesel engine.
The Rockford clutch has a balanced toggle-action which holds the clutch firmly in the engaged or the disengaged position without running thrust, and a simple and accessible device which is self-locking without the need for special tools.
British made, in a range of sizes, by
BORG & BECK COMPANY LTD.,
LEAMINGTON SPA, ENGLAND
ROCKFORD
POWER TAKE-OFFS & CLUTCHES




LOCKHEED
On the
STANDARD
VANGUARD


ON THE VAUXHALL
Thompson
Self-adjusting
STEERING ROD ASSEMBLY
AUTOMOTIVE PRODUCTS COMPANY LIMITED
LEAMINGTON SPA, ENGLAND

Red-hot ash in a timber truck...

Ash at a dull red heat is discharged directly into this lorry from the collectors of a producer plant and carried to the tipping point. After a year and a half of this work the wooden sides are in perfect condition and even the paint shows no discoloration . . .

Incredible? . . . Well we haven't told the whole story, of course. There's half-an-inch of 'Fibreglass' between the ash and the timber. This insulation covers the whole of the bottom and sides and is protected by thin mild-steel sheet on the inner side. Just another example of how heat can be kept in its place...

...thanks to FIBREGLASS

TRADE MARK

For structural, heat and cold insulation. Sound-deadening. Acoustic correction. Porous membranes for pipe wrapping, flooring, roofing. Battery separators and air filters. In textile form for electrical insulation and flameproof decorative fabrics.

FIBREGLASS LTD., RAVENHEAD, ST. HELENS, LANCS. (ST. HELENS 4224)

LONDON OFFICE: *63/65 Piccadilly, W.1 (Regent 2115/6)* **GLASGOW OFFICE:** *136 Renfield Street (Douglas 2687)*
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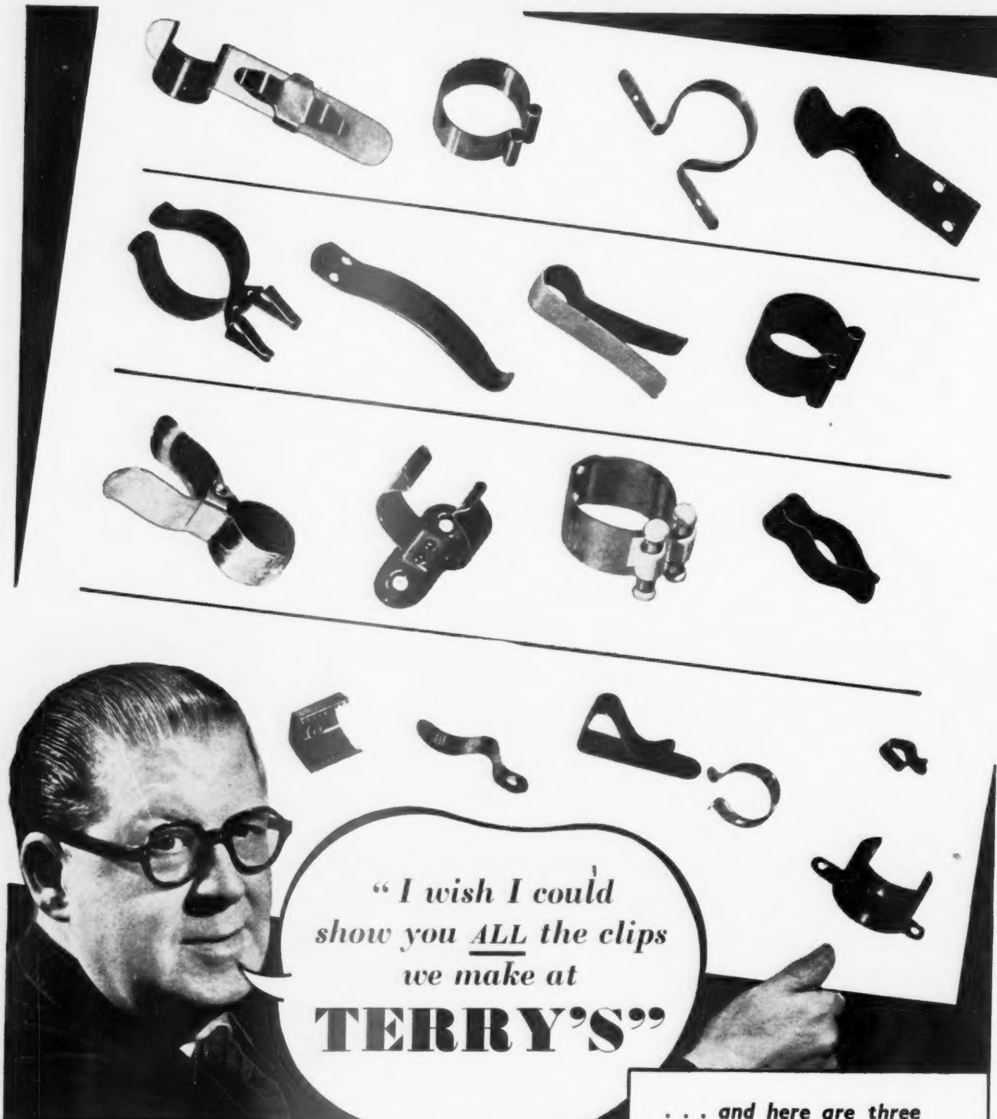
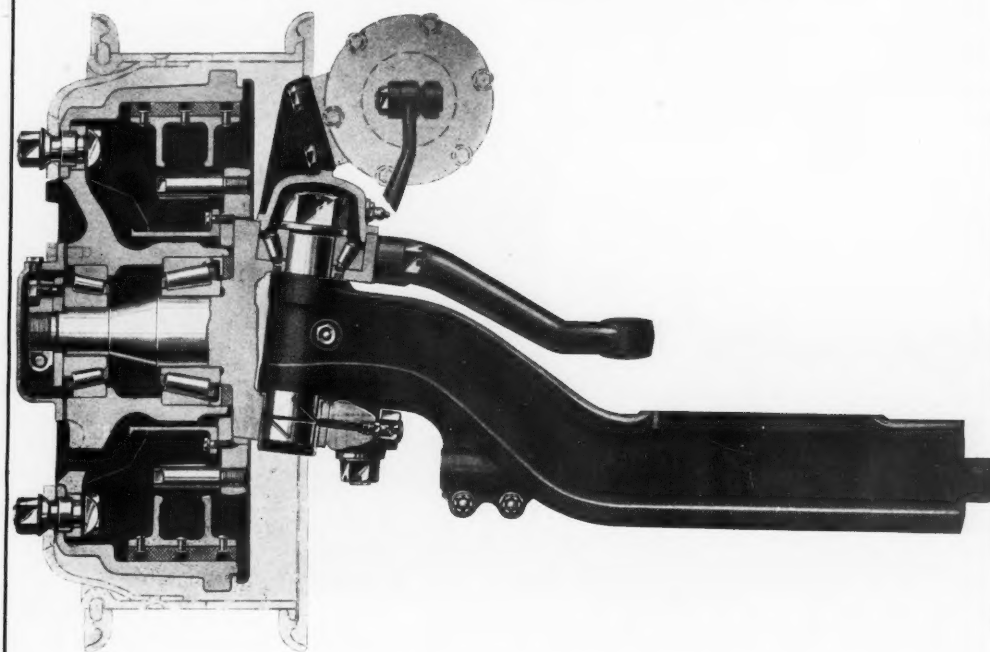
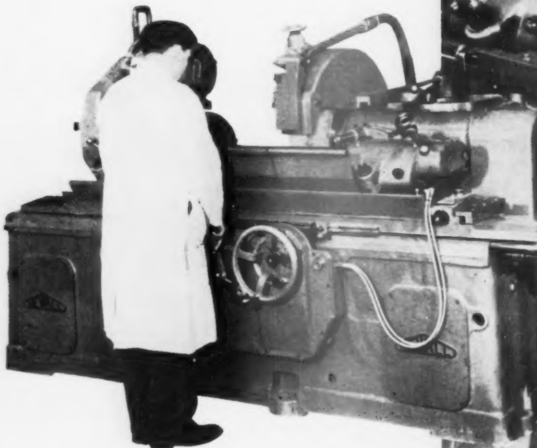
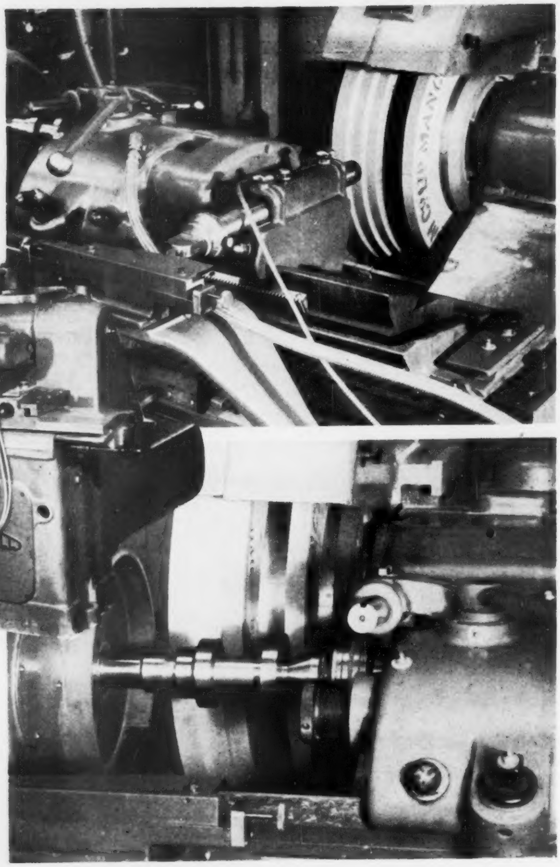


"I wish I could show you ALL the clips we make at TERRY'S"
HTSA





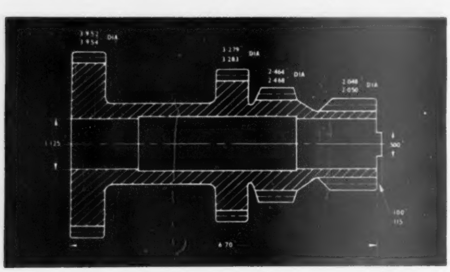


This component, a steel forging hardened to Rockwell 58° Scale 'C,' was plunge ground on four diameters in only 8 seconds, whilst the redressing of the grinding wheel was accomplished in 5 minutes.

★ *Actual grinding time 8 secs!*

Grind all diameters at one plunge..

In common with other leading automobile manufacturers, the Ford Motor Co., (Canada) Limited employ only the most modern production methods. Amongst their tools is this Newall 10″ × 24″ L.A. Grinding Machine. With this machine, all diameters of Gear Transmission Countershafts can be ground at one plunge—the actual grinding time taken being only 8 seconds. The L.A. Grinding Machine has proved to be the right tool for high and economic outputs.



◀ *gear transmission countershaft*

NEWALL GROUP SALES LTD.
PETERBOROUGH Tel. PETERBOROUGH 3227/8/9
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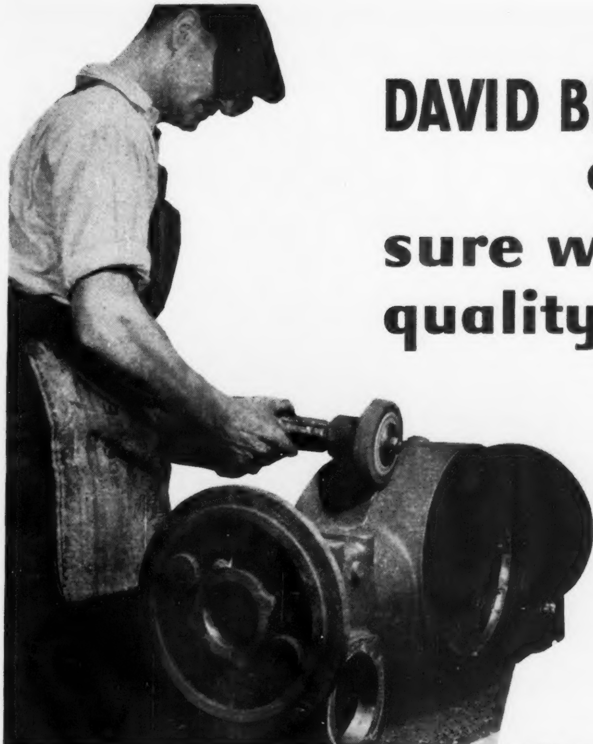


Another testing machine from the Bury Felt laboratory. This stretches the selected sample of felt, testing its behaviour under strain and its conformity with the predetermined breaking point.

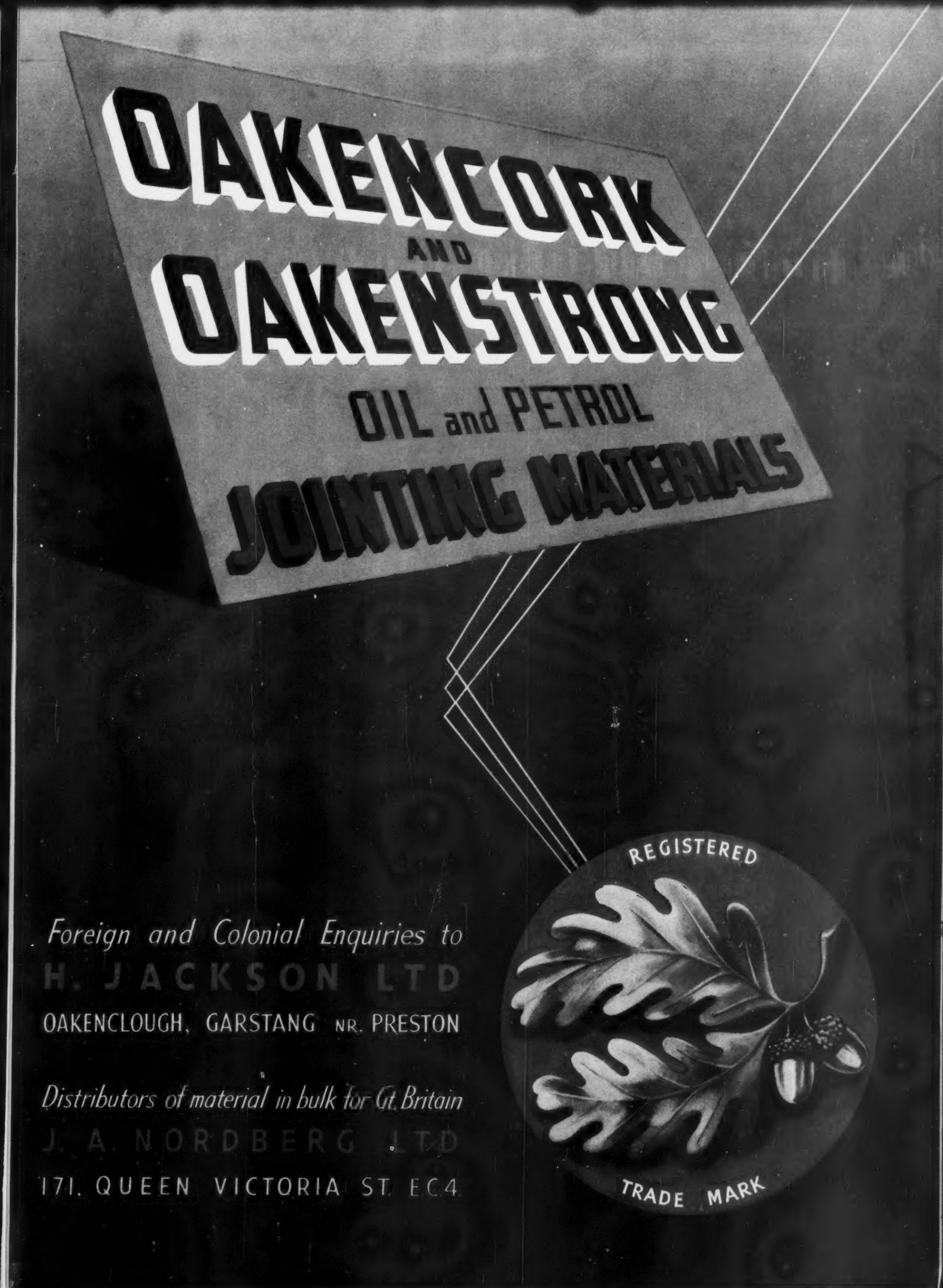
Felt, like many other industrial products, must measure up to precise standards of behaviour under particular stress. The Bury Felt laboratory, with its special testing equipment, ensures that all Bury Industrial felts are ready to meet the particular needs for which they have been made.

In filters, seals, washers, buffing rollers, shock absorbing mountings and cushionings Bury Felts give a good account of themselves. These are just a few uses of the versatile materials which can be die-cut, chiselled, punched machined and even ground.

BURY FELT

Send your enquiries to:

BURY FELT MANUFACTURING COMPANY LIMITED
HUDCAR MILLS, BURY, LANCS; Phone: Bury 2262 (6 lines)
or to the London Office: 3 Snow Hill, E.C.1. Telephone: Central 4448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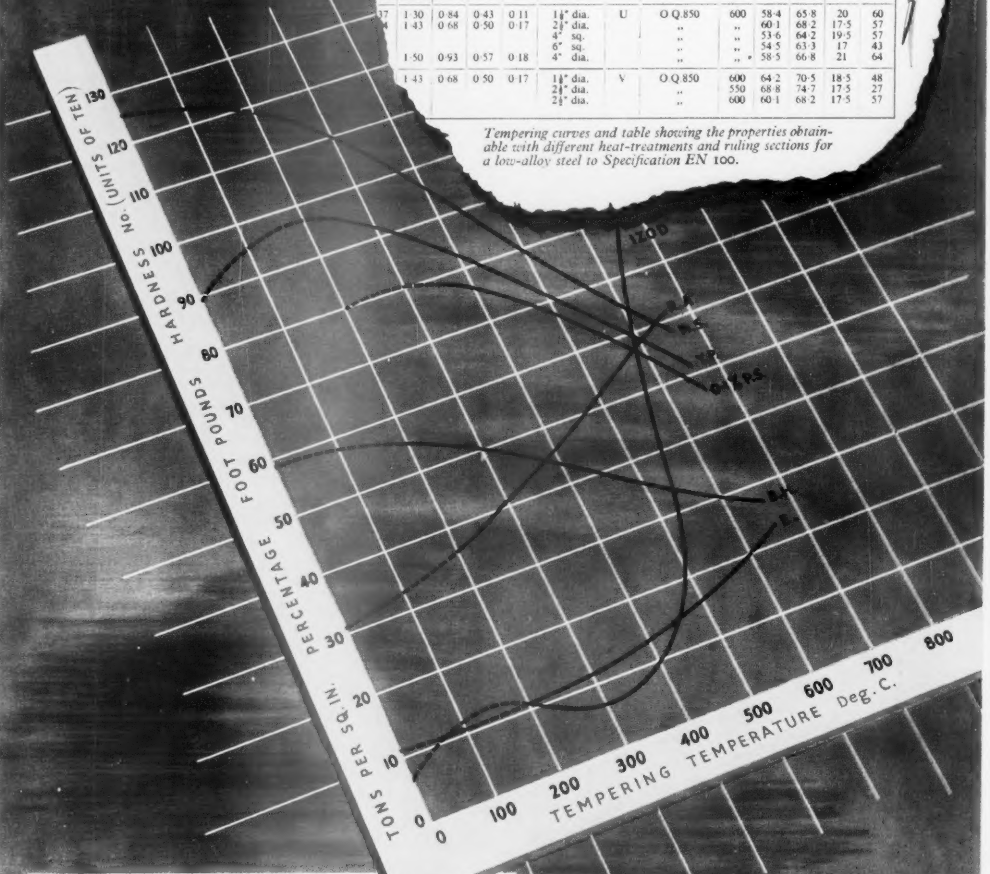
171, QUEEN VICTORIA ST. E.C.4

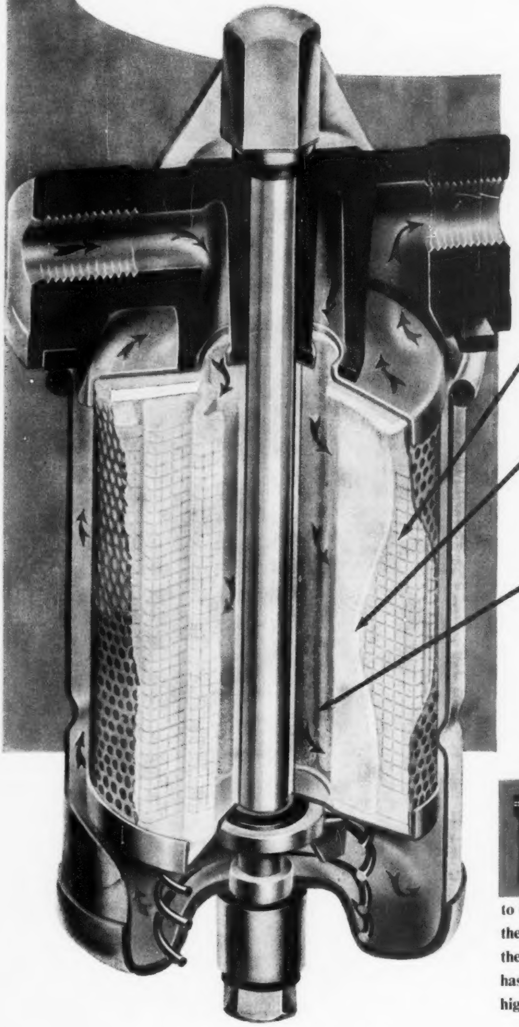
At Ebbw Vale: checking temperature of steel in Bessemer converter by immersion pyrometer.

		Ni	Cr	Mo	Size inches	Tensile Range Symbol	Heat-Treatment Harden C.	Temper C.	Mechanical Test Y.P. t.s.i.	M.S. t.s.i.	Elong per cent.	Izod ft.-lb.
		0·73	0·48	0·18	2½" dia.	S	O.Q.850	650	51·2	58·2	22	81
					5" sq.		,,	,,	43·1	53·4	17·5	68
7		0·73	0·49	0·17	6" sq.		,,	,,	46·8	56·9	21·5	80
·28		0·74	0·46	0·26	6" sq.		,,	,,	46·0	56·2	21·5	75
	1·50	0·93	0·57	0·18	2½" dia.		,,	,,	44·8	55·2	21·5	89
					4" dia.		,,	,,	48·0	57·2	24	87
39	1·41	0·90	0·38	0·18	4" sq.		,,	,,	46·8	57·0	22	69
35	1·50	0·93	0·57	0·18	2½" dia.	T	O.Q.850	600	52·2	62·8	19	43
					4" dia.		,,	650	48·0	57·2	24	87
39	1·41	0·90	0·38	0·18	2½" sq.		,,	,,	51·2	57·6	24	94
37	1·30	0·84	0·43	0·11	1½" dia.	U	O.Q.850	600	58·4	65·8	20	60
	1·43	0·68	0·50	0·17	2½" dia.		,,	,,	60·1	68·2	17·5	57
					4" sq.		,,	,,	53·6	64·2	19·5	57
					6" sq.		,,	,,	54·5	63·3	17	43
	1·50	0·93	0·57	0·18	4" dia.		,,	,,	58·5	66·8	21	64
	1·43	0·68	0·50	0·17	1½" dia.	V	O.Q.850	600	64·2	70·5	18·5	48
					2½" dia.		,,	550	68·8	74·7	17·5	27
					2½" dia.		,,	600	60·1	68·2	17·5	57

Tempering curves and table showing the properties obtainable with different heat-treatments and ruling sections for a low-alloy steel to Specification EN 100.



ARE YOU GETTING *FILTRATION IN DEPTH?*

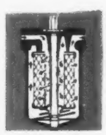


Why has the basic principle of the VOKES filter element remained unchanged for over thirty years despite constant claims for cheaper and so called better methods? That is the vital question to be asked when considering filtration claims.

Here is shown the VOKES felt fabric element with its wire gauze outer retainer . . . not just filtration in only one plane . . . not just one surface through which particles can be driven.

Here is *filtration in depth* **giving greater efficiency in arresting all particles large enough to span the oil flow . . . giving a build-up of sludge much slower than would be the case with an element filtering on its outer surface only.**

Remember, also, the fact that in the VOKES type, filtration is from the inside to the outside which means that all the sludge is trapped in the *centre* **of the filter element and cannot, in any circumstance, be allowed to recirculate.**



You must filter all the oil all the time—even a full flow filter is not doing its job if the by-pass is brought into use for long periods when the oil is only ordinarily cold. The VOKES direct flow device exists purely as a protection to the engine and not to the filter. It only allows the oil to pass straight through to the engine if the filter has been so neglected that the element has been completely clogged up or if the oil is highly viscous through abnormal conditions.

★ ***VOKES FABRIC ELEMENTS ARE CLEANABLE!***

They give greater economy in the long run and ensure that plant and machinery is not put out of service through the non-availability of spare elements in remote places.

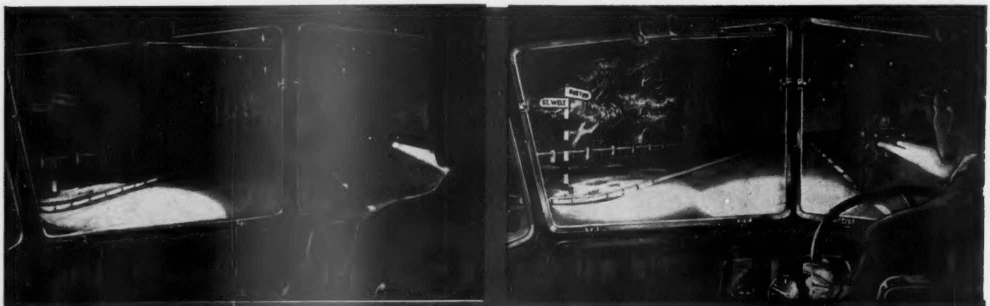
VOKES

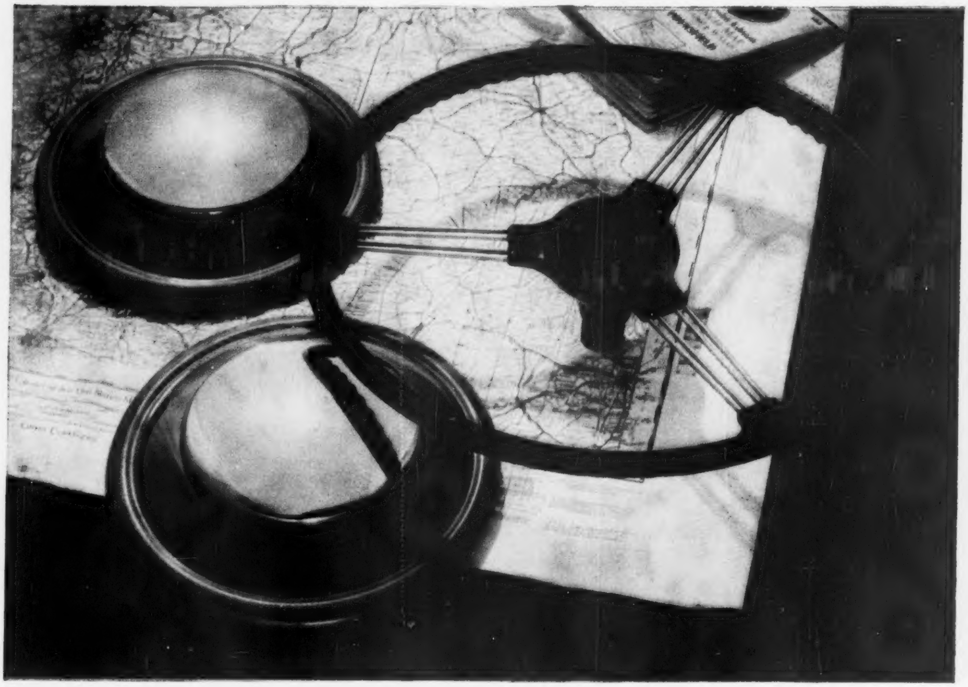
Pioneers of scientific filtration

V63

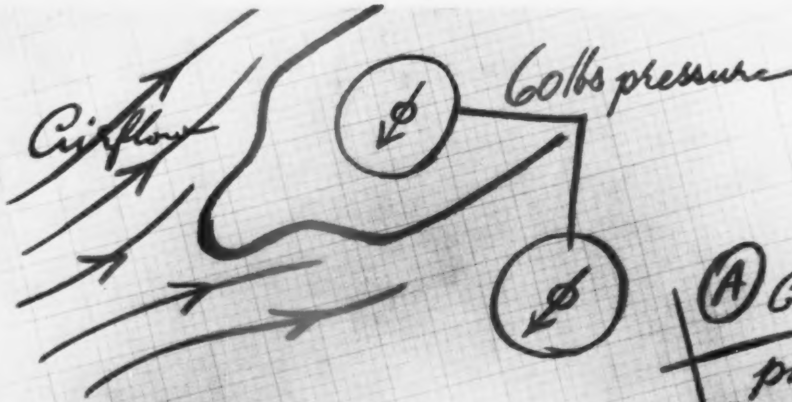
VOKES LIMITED GUILDFORD SURREY

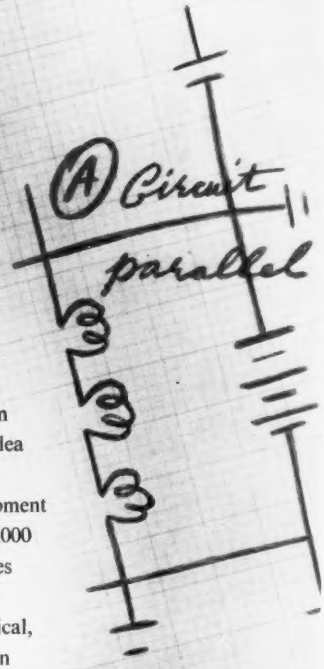
VOKES (CANADA) LTD. Toronto — *Represented Throughout the World* — VOKES AUSTRALIA PTY LTD. Sydney





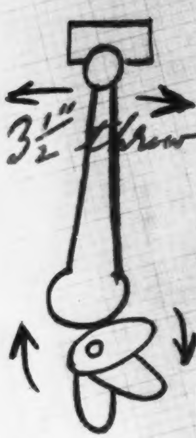
P.P.I

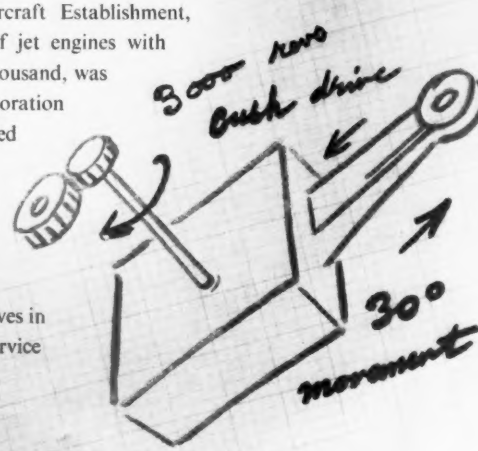




MAKING DOODLES WORK

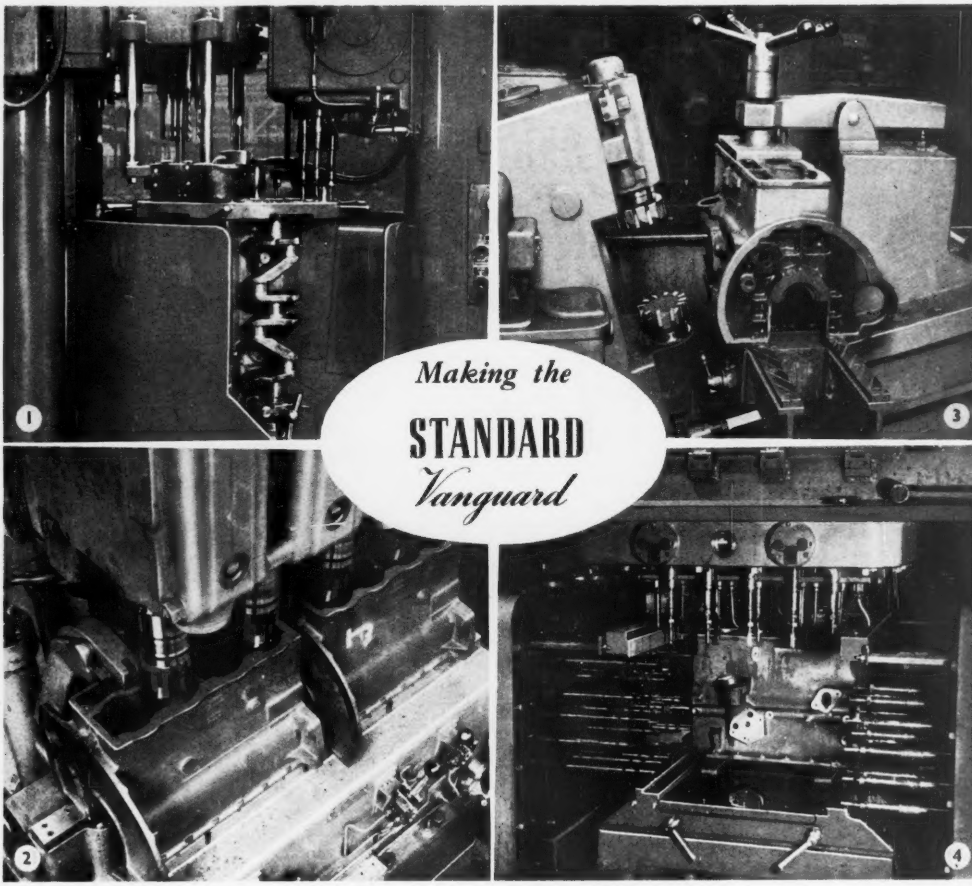
Pass the rough outline of something new and how it must perform to Plessey, who will develop, test, and produce a prototype—your idea brought to life with shrewd refinements. *And* it will be easy to manufacture, in quantity, cheaply. That is Plessey design and development in simple terms. It can also be expressed in figures—more than 10,000 workers, over twenty acres of design offices, laboratories and production equipment. Or, if you remind us that capacity needs to be mental as well as physical, take this example: the Electronic Revolution Counter at the Royal Aircraft Establishment, which measures the speed of jet engines with an accuracy of one in a thousand, was developed in collaboration with RAE and produced throughout by Plessey. Now you have some idea of the unique combination of technical brain and mechanised brawn at your command. Executives in industry or Government service may like us to explain how...





Plessey PUTS IDEAS INTO PRODUCTION

THE PLESSEY COMPANY LIMITED · ILFORD · ESSEX



1. CRANKSHAFTS.
Drilling, counterboring and tapping four $\frac{3}{8}$ in. by 24 T.P.I. holes in flange, drilling and reaming two 0.3748/52 in. dowel holes, $\frac{5}{16}$ in. deep and continuing through flange at $\frac{3}{16}$ in. dia.

2. CYLINDER BLOCK.
Two components loaded in tandem, "through" loading. First and third bores for cylinder liners at one position, and second and fourth bores for cylinder liners at the second. position, rough bored and faced at one pass of machine head

3. CYLINDER BLOCKS.
Special duplex angular milling machine simultaneously milling the distributor and pump faces.

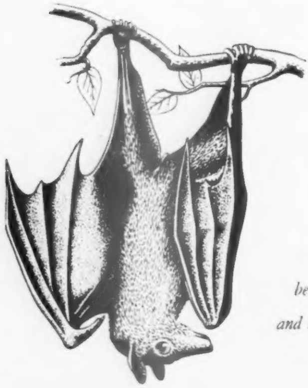
4. CYLINDER BLOCKS.
Special three-way machine tapping the head joint face and the two end faces of cylinder blocks. Each spindle has individual lead screw control.

ARCHDALE *SPECIALS*

for high production in the automobile industry

JAMES ARCHDALE & CO. LTD., LEDSAM STREET, BIRMINGHAM, ENGLAND

SOLE SELLING AGENTS: ALFRED HERBERT LTD., COVENTRY & BRANCHES

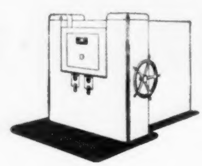


One of the easiest ways of getting a headache is to hang upside down—at least most people think so—but the Bat seems to like it. Maybe it's because he's as blind as a bat and doesn't know the difference. Good job he doesn't have anything to do with heat treatment. There is a world of difference between equipment that will serve to do the job, and that which is specially designed to do it.

perfectly designed to do the job

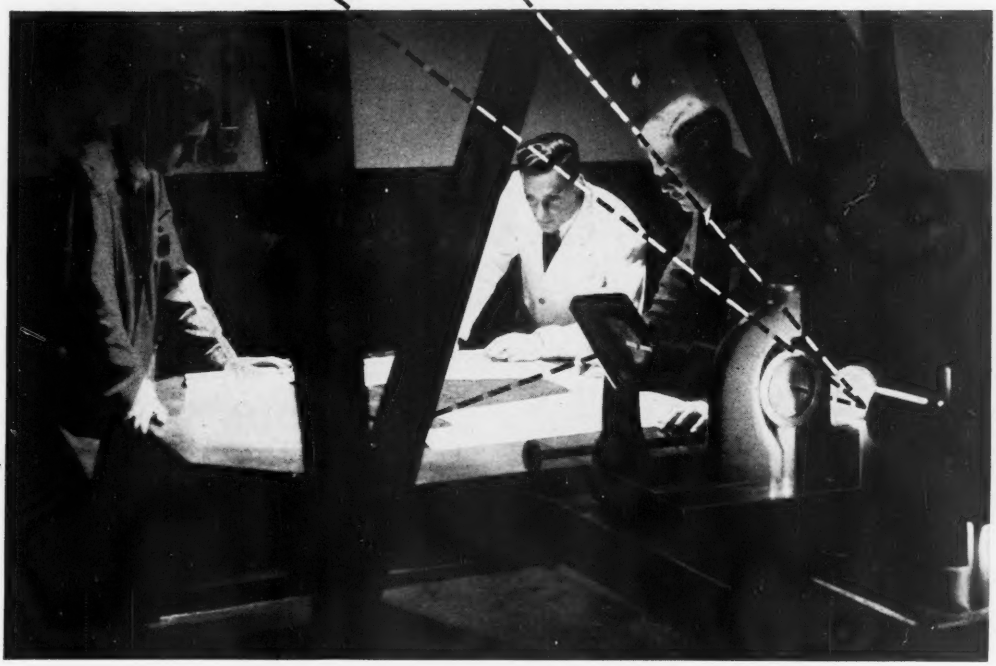
The Wild-Barfield Heavy-Hairpin Furnace is designed to do the job too. With a temperature range of 700° C. to 1050° C. (or with "Tubular-Hairpin" elements up to 1150° C.) they are suitable for the majority of batch type heat-treatment processes. They are specially made for the job of hardening, carburising, normalising, annealing, and vitreous enamelling.

Wild-Barfield
Electric Furnaces



FOR **ALL** HEAT-TREATMENT PURPOSES

WILD-BARFIELD ELECTRIC FURNACES LTD. ELECFURN WORKS WATFORD HERTS. PHONE: WATFORD 6094 (4 LINES)



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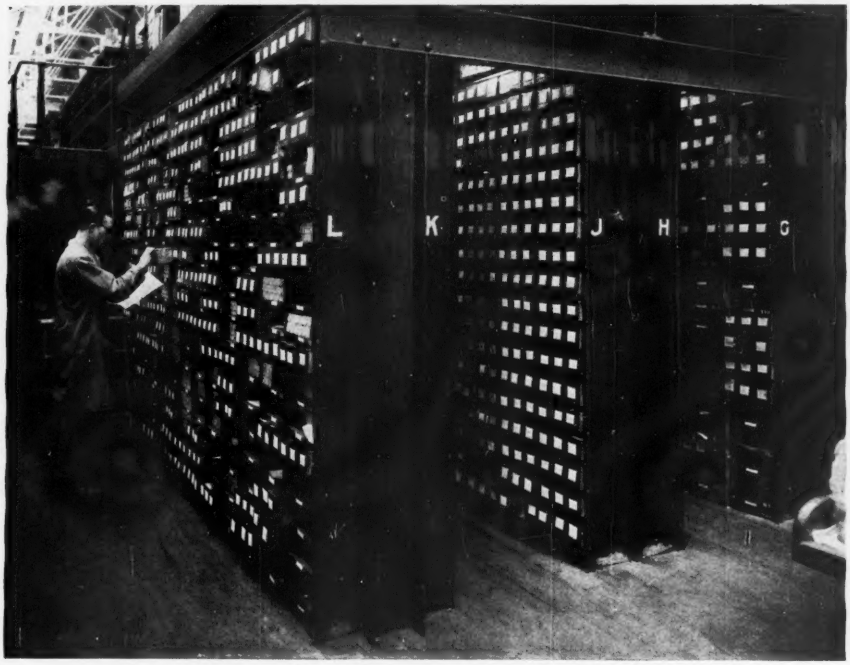

COVENTRY DIES IN STOCK

WE have a large carefully selected stock of Coventry Dies, comprising all sizes and types in general use. Orders for these will be executed immediately.

Other dies can be delivered in 6 weeks from receipt of order.

Coventry double-tempered dies are made in ten types to suit the different materials now in use, including the alloy steels that are difficult to cut.

We guarantee to supply dies which will cut clean, accurate threads on any material that can be threaded.

PLEASE SEND ENQUIRIES TO THREADING DEPT.
TELEPHONE 88781. EXTENSION 210.

URGENTLY REQUIRED
Capacity for surface and form grinding of carbide-tipped form tools.

ALFRED HERBERT LTD · COVENTRY


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HIGH PRESSURE flexible HOSE
lifeline of your
ucts
Your machines may be of best quality
the highest standard
aftsmanship, but without a perfect
can all be rendered
Hyline High Pressure
ensures constant flow to
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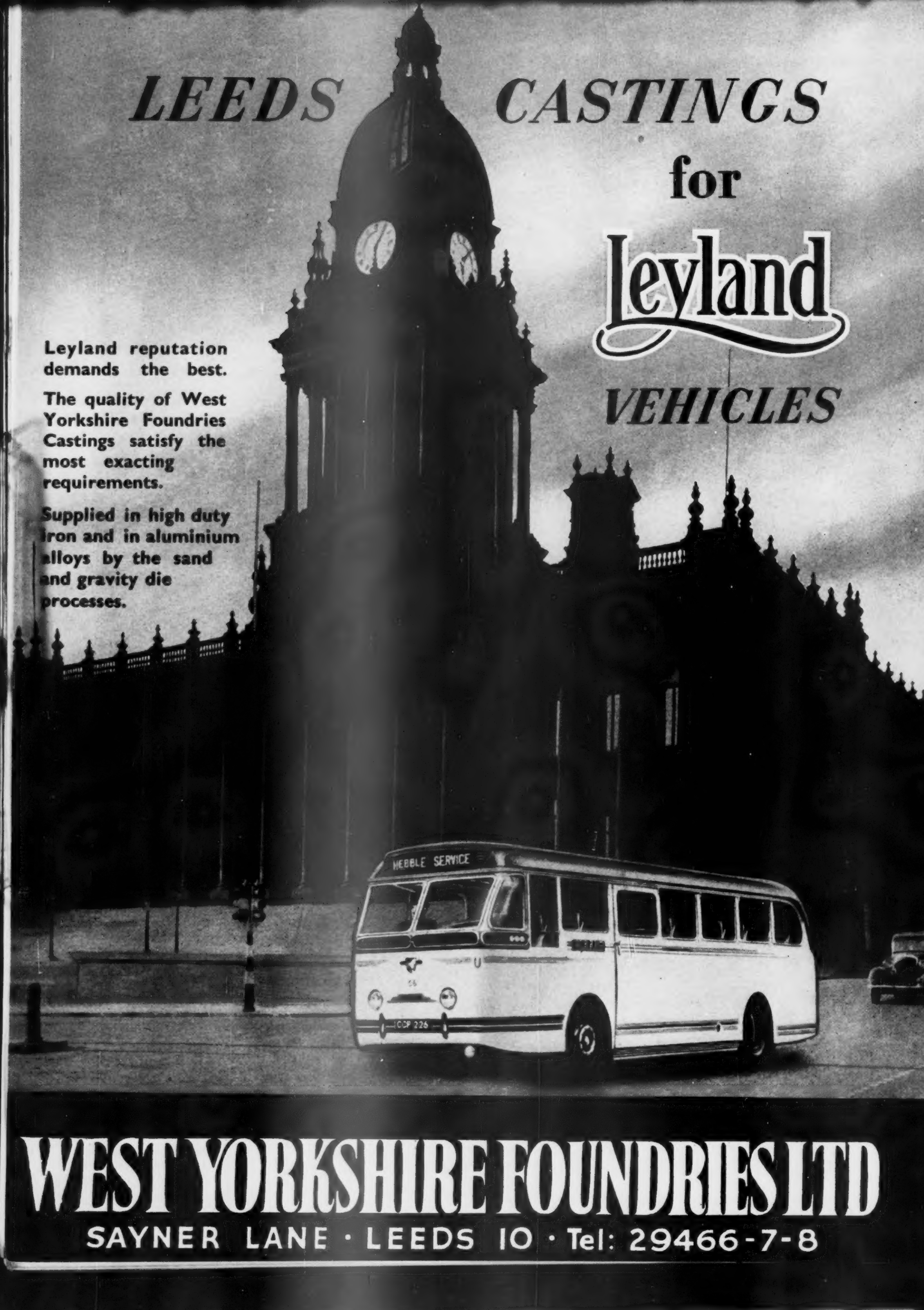
vital parts
machine at the peak of
efficiency.
liquids and greases at a wide
range of
great pressures. Supplied in lengths up
to 30 feet and in a range of bores; complete with end fittings.
A SUPERFECT PRODUCT
SUPER OIL SEALS & GASKETS LIMITED
FACTORY CENTRE
BIRMINGHAM 30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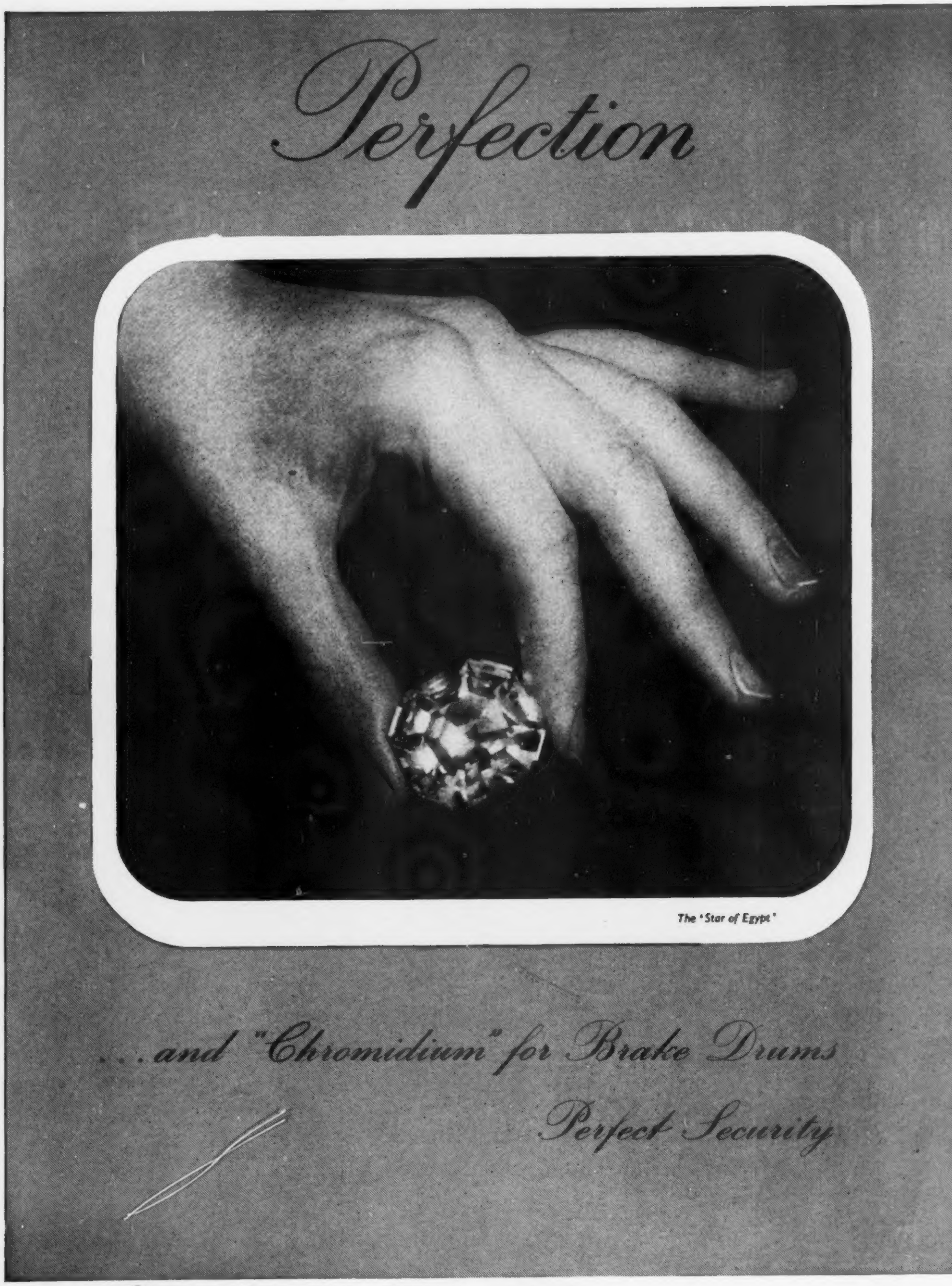
Perfection
The 'Star of Egypt'
...and "Chromidium" for Brake Drums
Perfect Security


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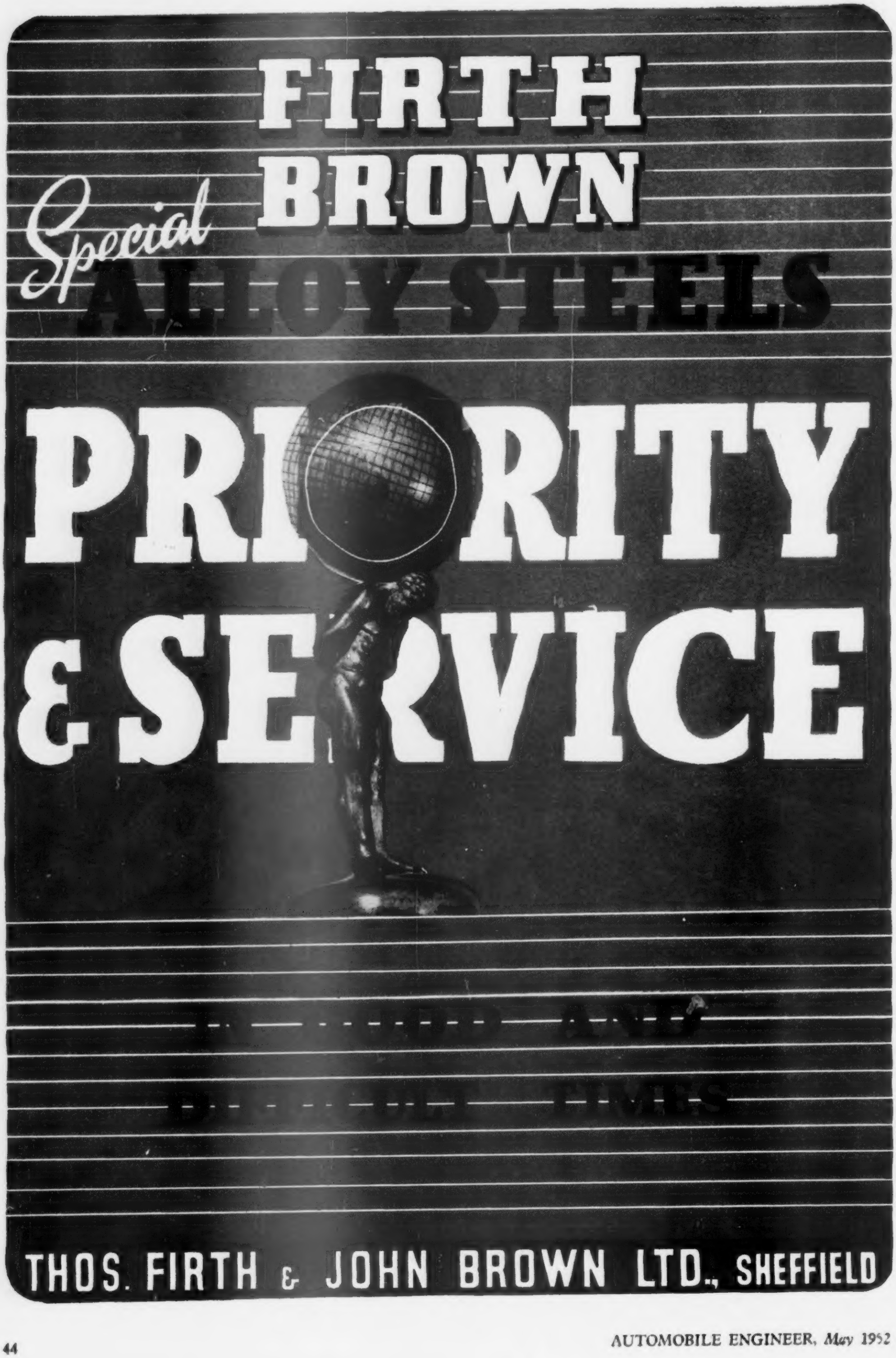
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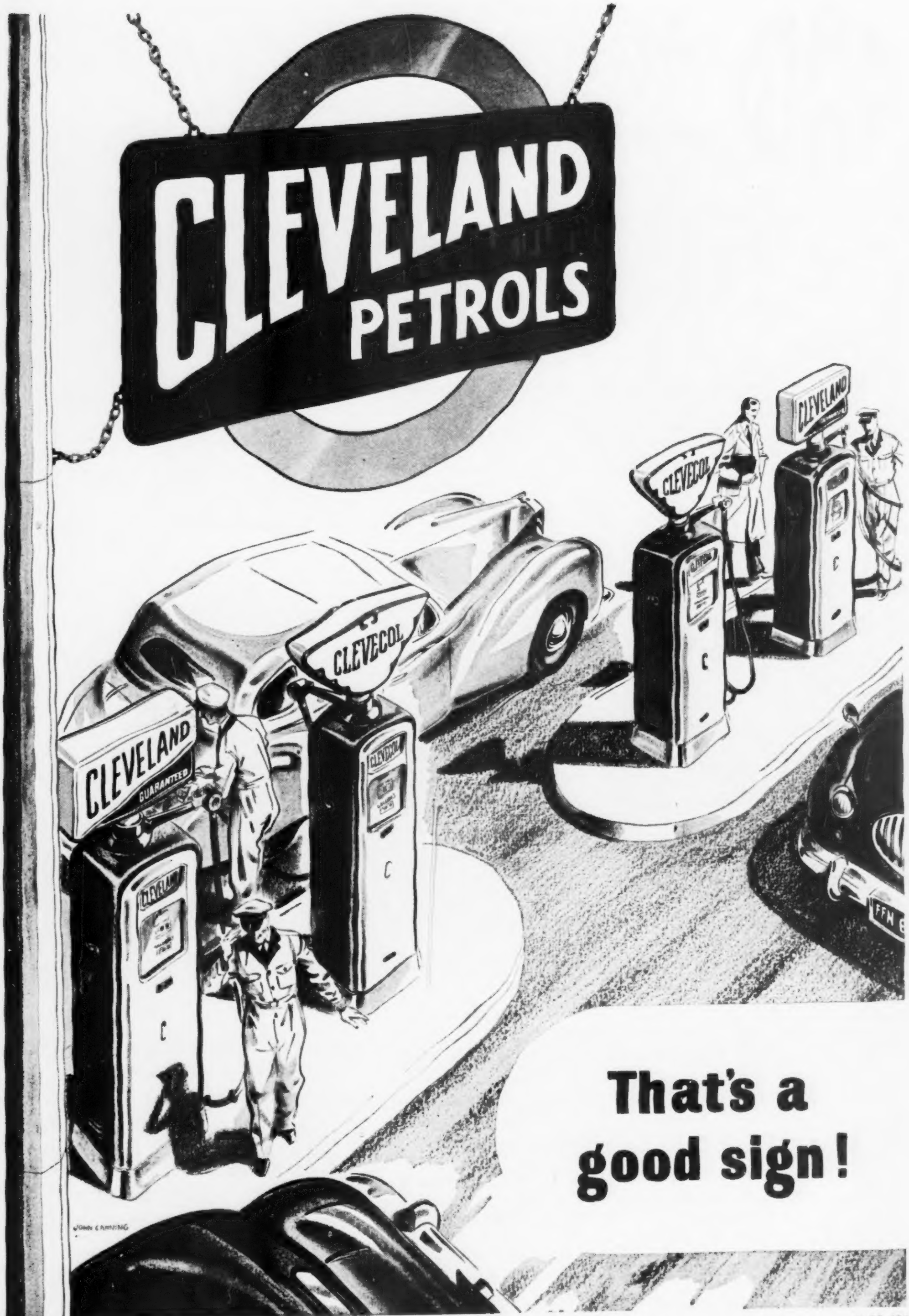
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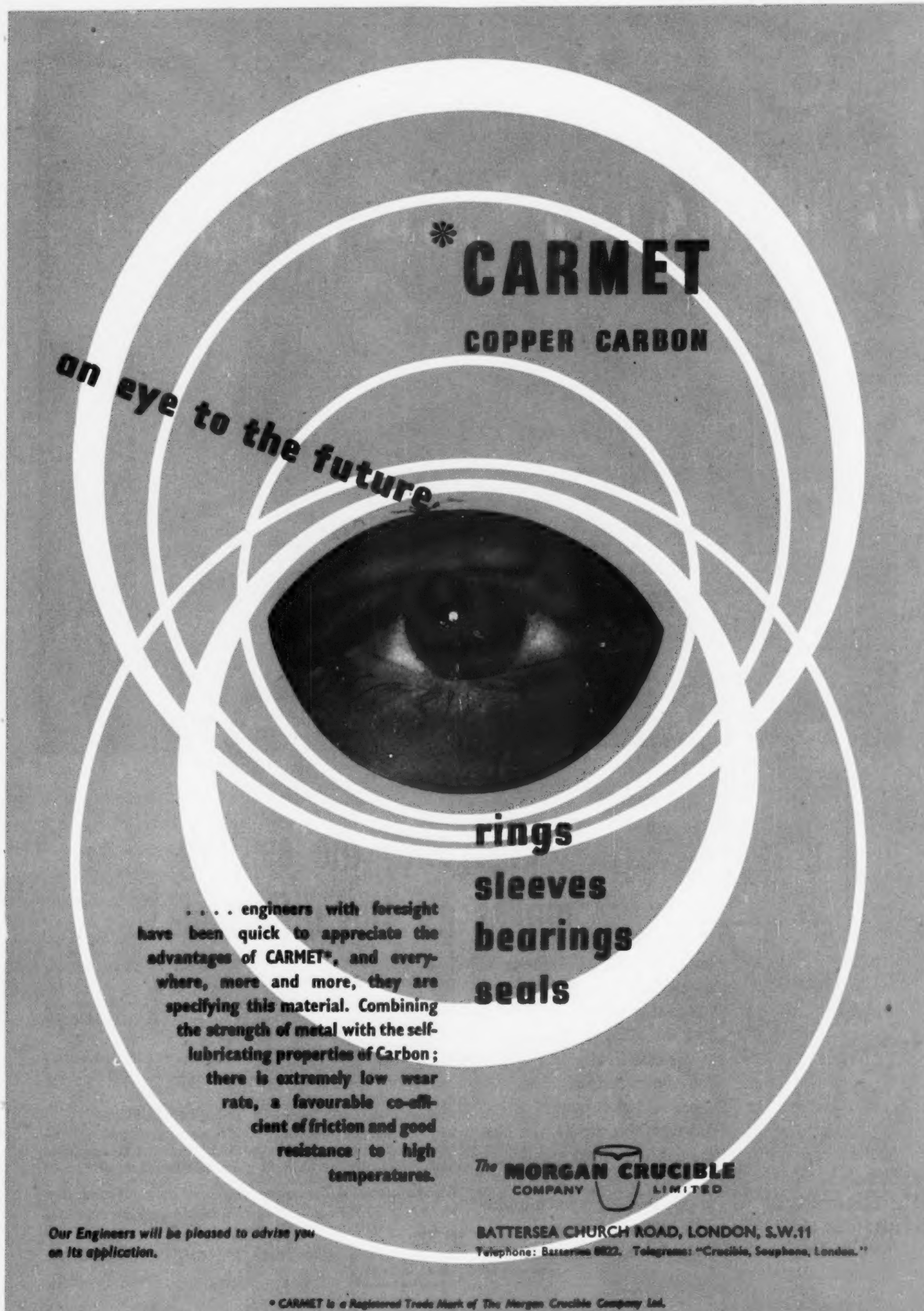
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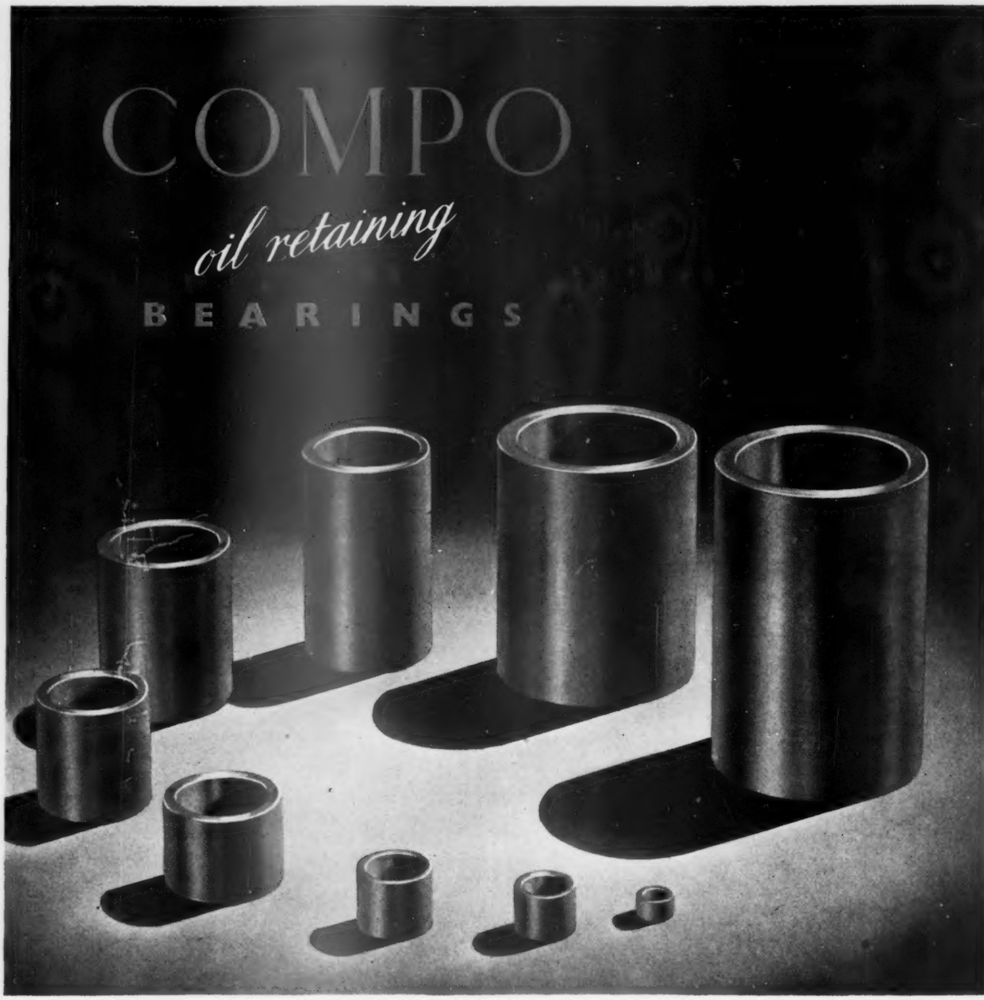

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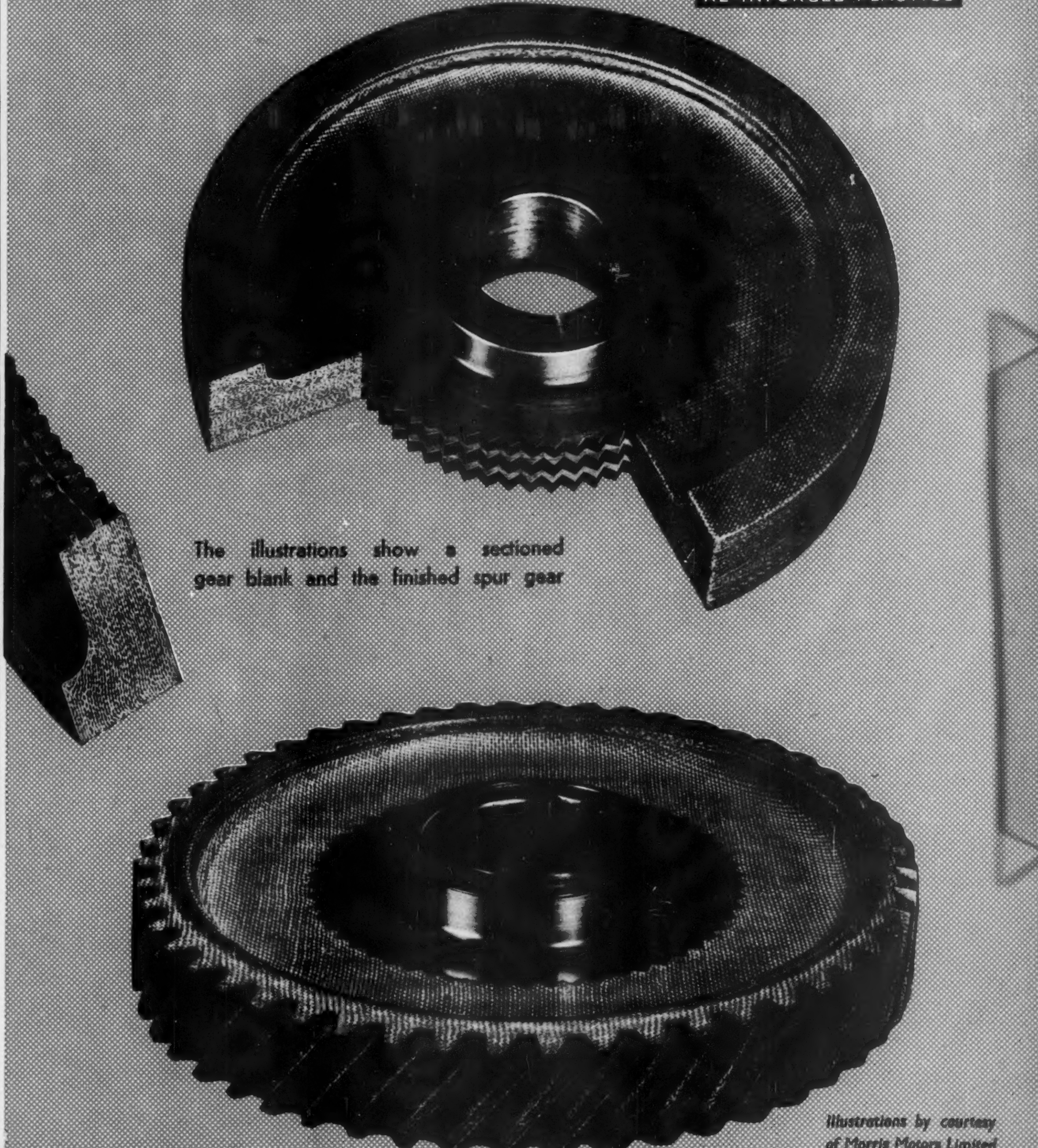

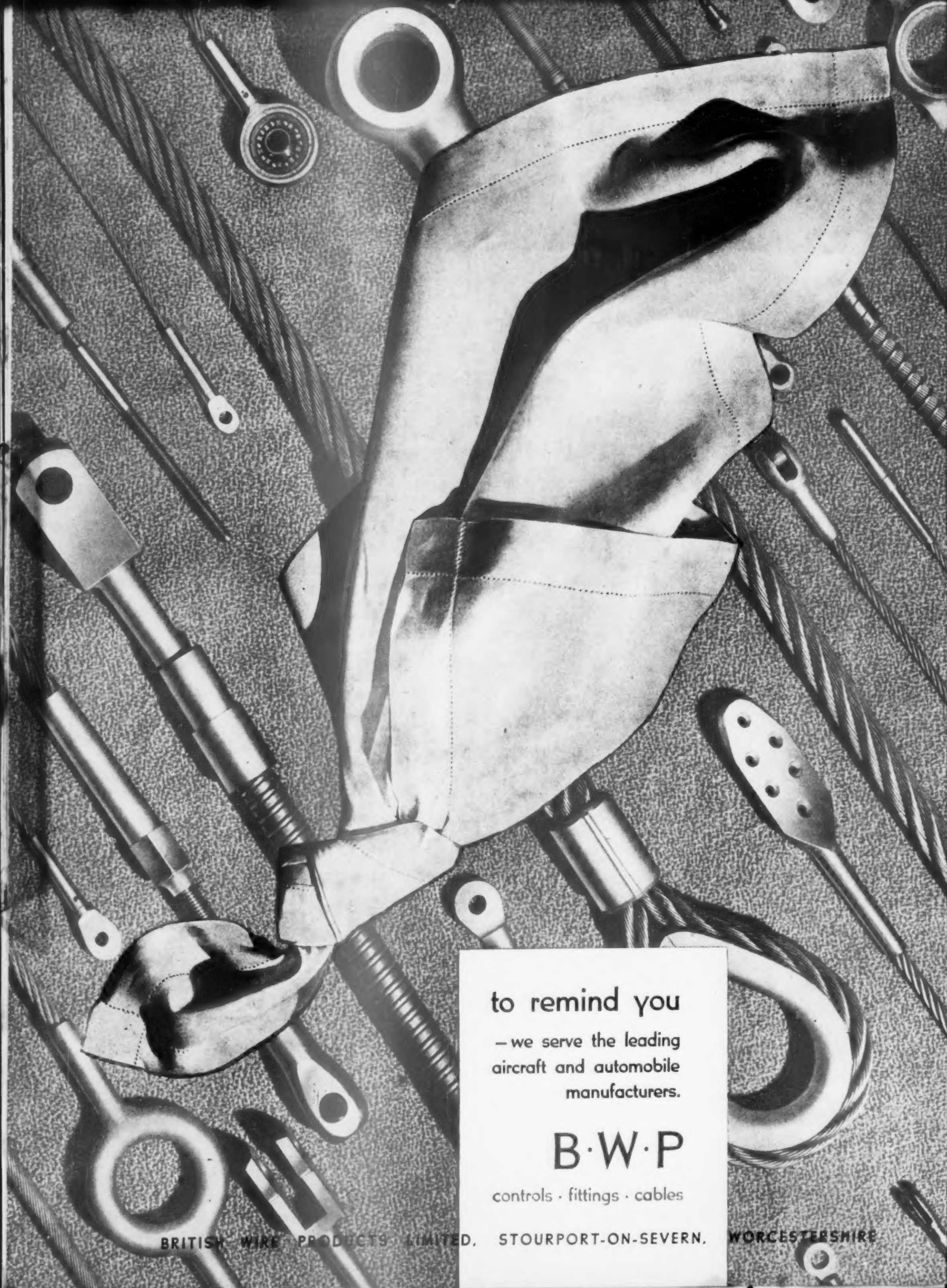
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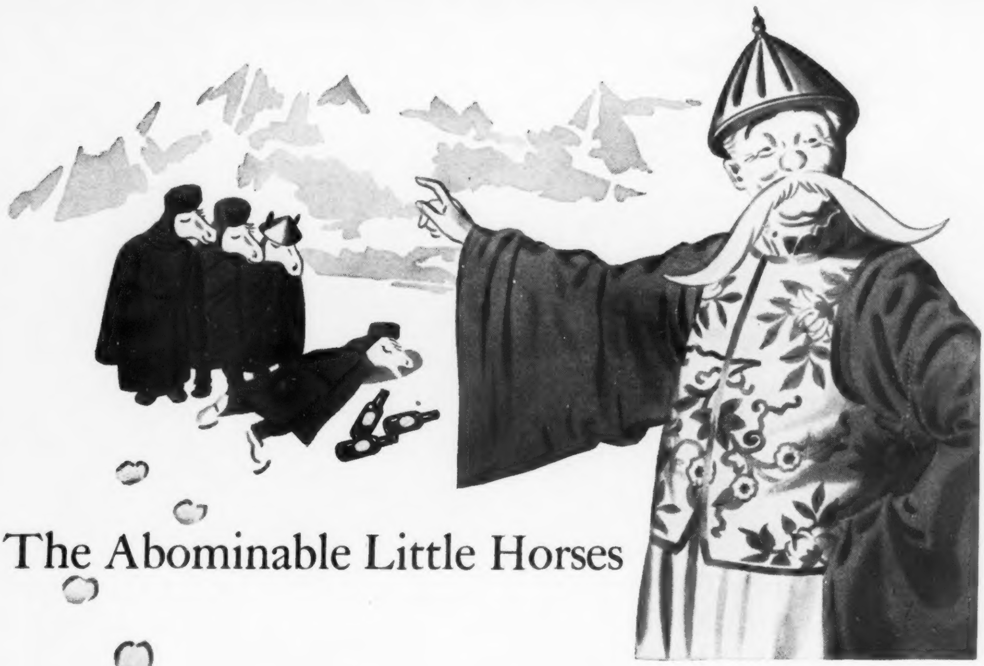

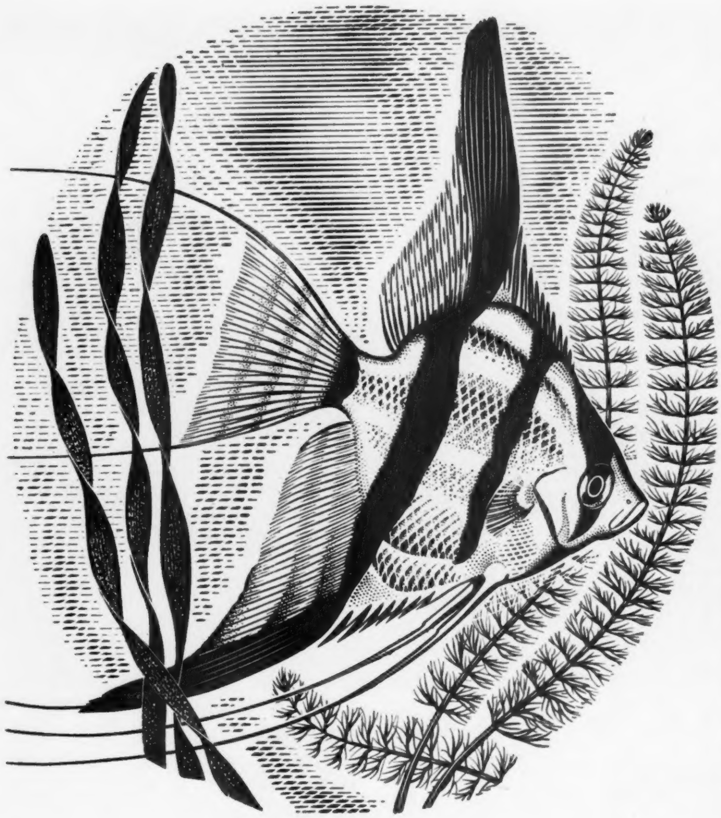

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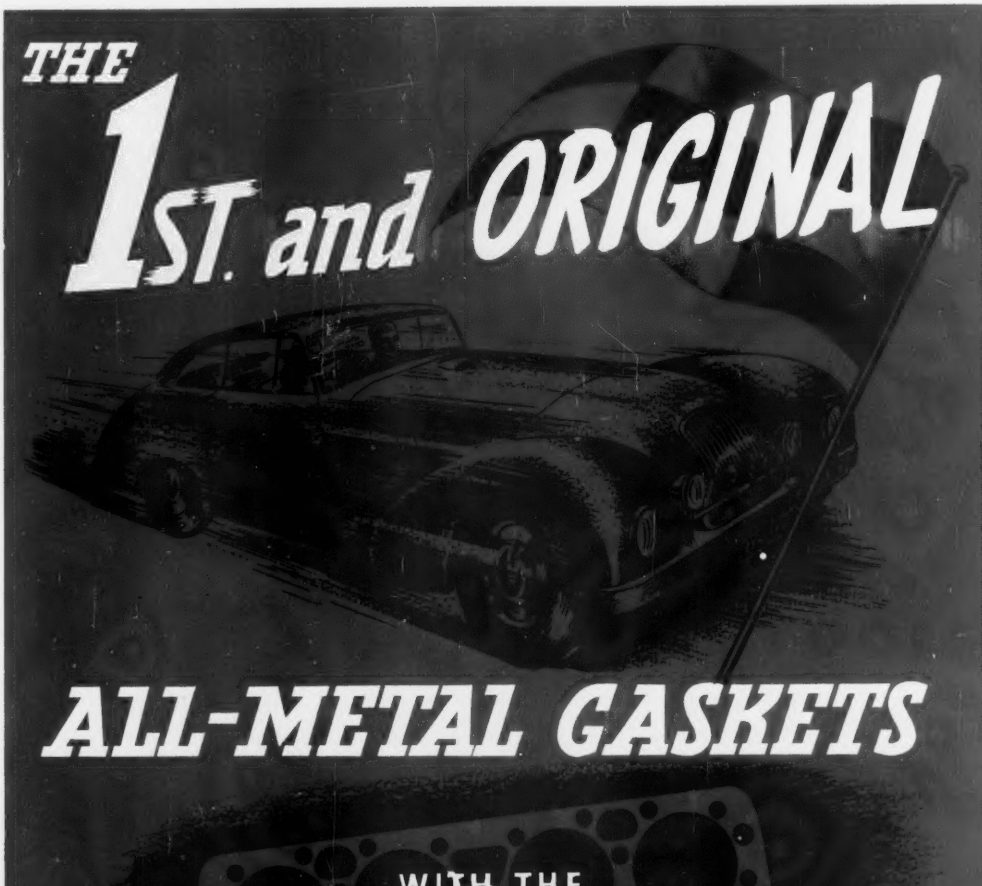
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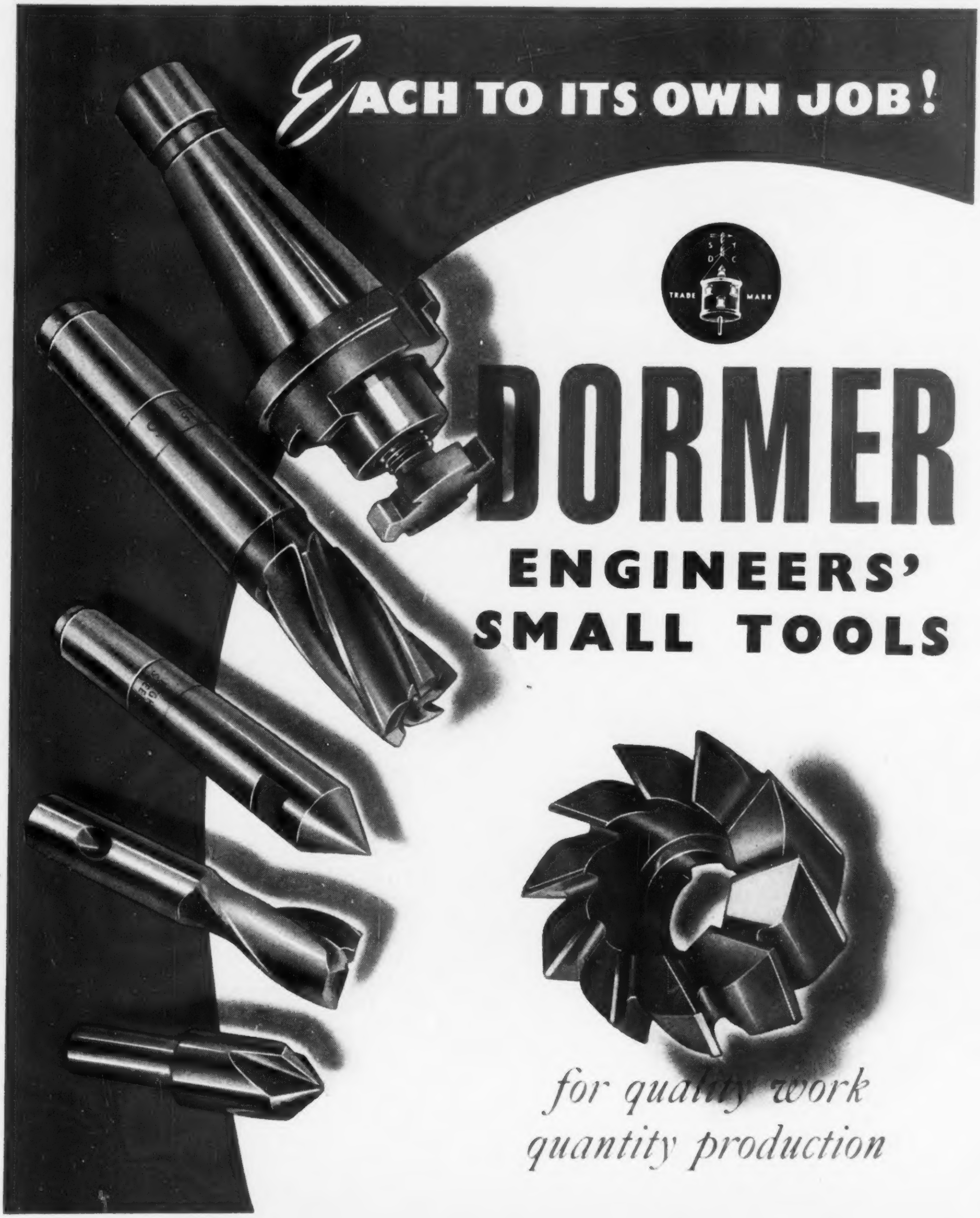
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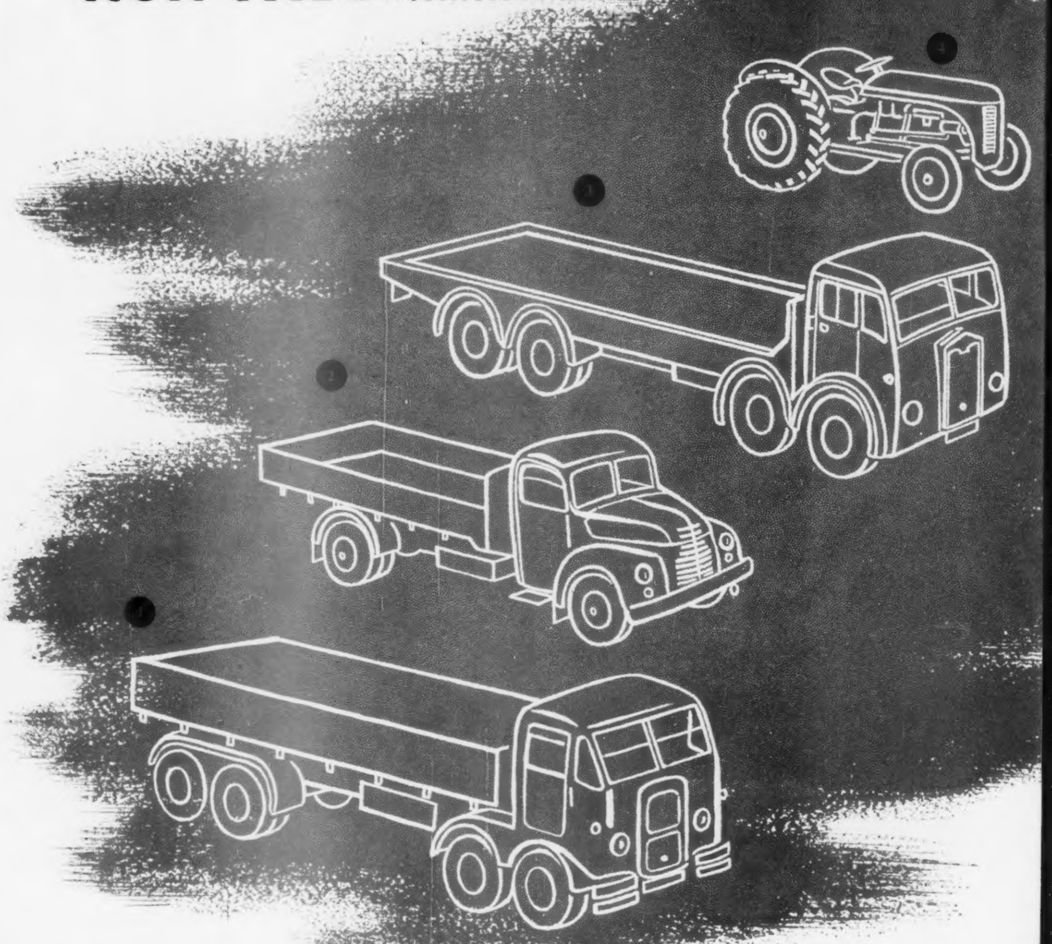

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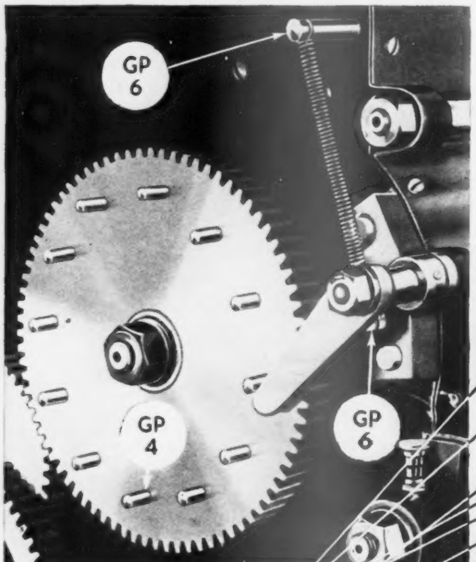

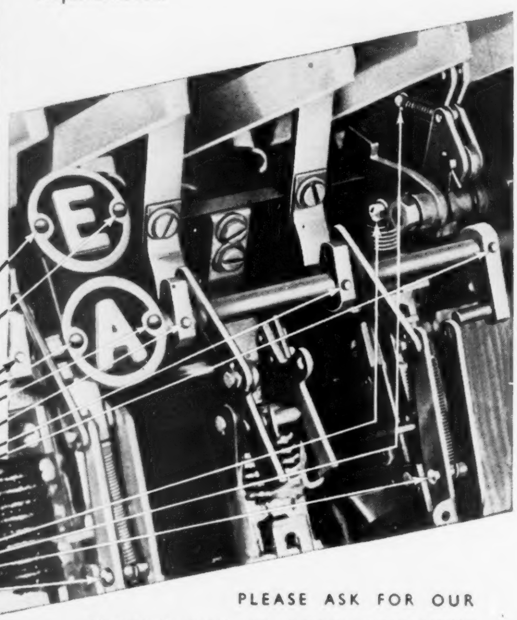

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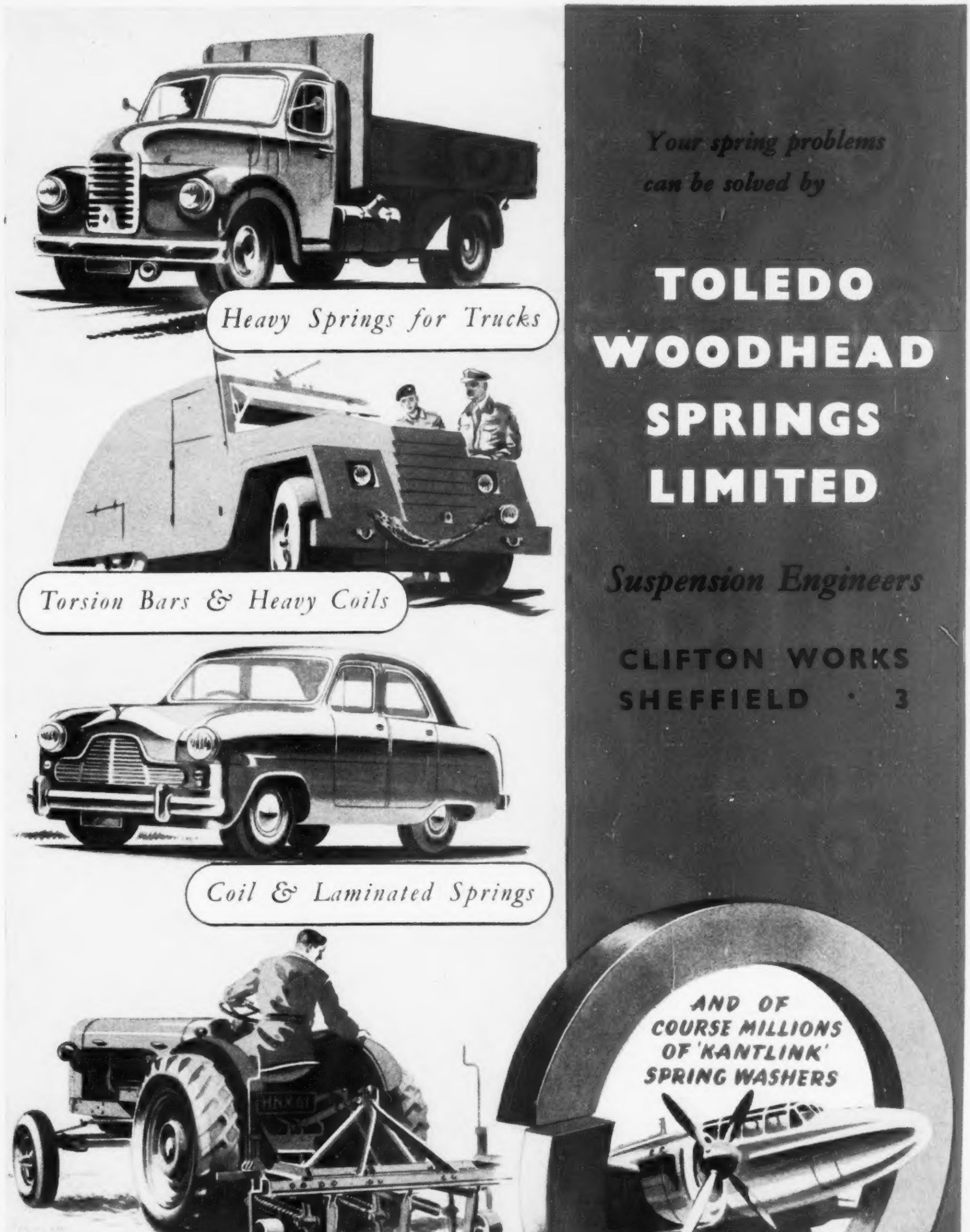
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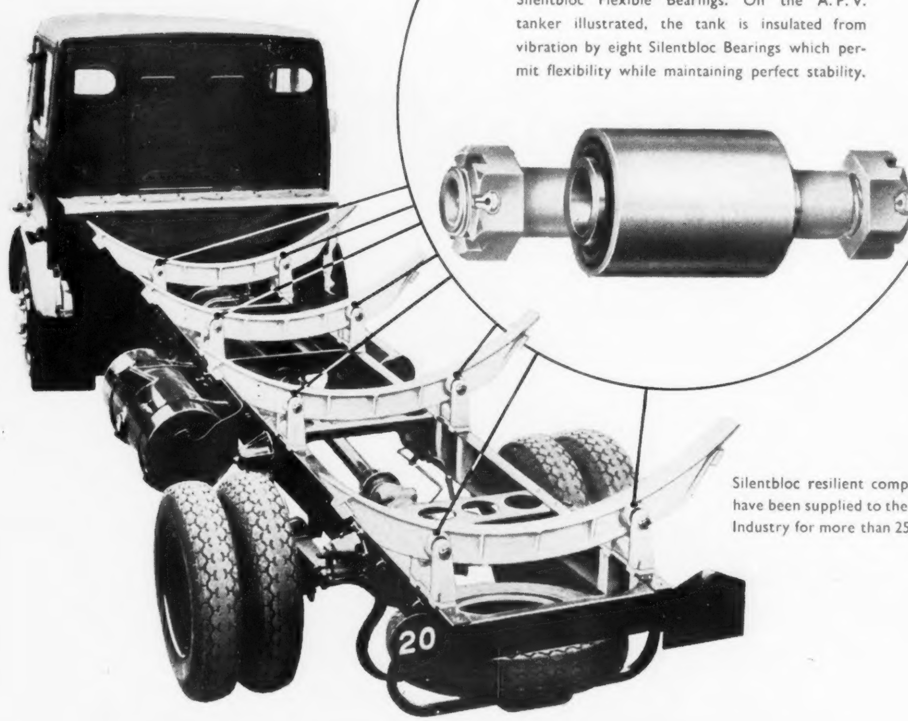

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Materials Handling

SINCE precise terminology is a pre-requisite of exact thinking, it is unfortunate that the term "mechanical handling" is in much more frequent use than the term "materials handling". The former refers to only a part, albeit an important one, whereas the latter embraces the whole of a subject that is of fundamental importance in industrial economics. In saying this we do not intend to decry the importance of mechanical handling systems, but we do wish to emphasize that optimum efficiency can be obtained only when the problem of handling materials from the moment they enter the factory until the finished products are despatched, is considered as an integral whole. We may even go farther and say that co-ordination with material suppliers is necessary to complete integration.

In general, the problem has three components, storage, movement between sections or departments and movement through processes. At present, the second of these usually receives much more attention than the other two. In many organizations this may be justified, since it is there that the greatest, and we must add the most obvious, economies can be achieved. Nevertheless, even in such organizations, due weight ought to be given to the other components.

The primary aim of any mechanical handling system is, of course, to reduce transport costs to the lowest possible figure, and it is a matter of relatively simple accounting to determine what method of work movement from section to section or from department to department will give the lowest cost per unit moved. But another point should also be considered, but frequently is not. That is, the transport methods should be integrated with the storage and actual production functions to give the quickest possible turnover of materials.

Storage

The organization of an efficient materials handling system is far from being easy. It is in fact one of the most complex problems in many factories, much more difficult for example than the planning of a machining operation sequence. Many, and often conflicting, factors must be considered and given due weight. To take one example; the layout of a stores may present these alternatives. One will use comparatively little floor area and high stacking, while the other will require considerably greater floor area with relatively low stacking. Decision on this point does not affect only the superficial area reserved for stores; it may well be decisive in determining whether fork-lift trucks or low-loading trucks are the more suitable for moving materials in to and out of the stores.

Storing methods are also important. First-in, first-out should be the invariable rule. This may seem to be a statement of the obvious. Nevertheless, it is unfortunately true that there are still far too many stores in which issue of material is haphazard. Generally, these are organizations which for reasons of false economy use bins of excessive size, none of which is ever completely cleared of work before fresh parts are introduced.

Handling through operations

In the past few years progressive companies have shown much greater awareness of the importance attaching to the methods of handling work through an operation sequence. In the automobile industry, the best examples of efficient handling during an actual machining sequence are the multi-station automatic transfer machines that have been installed by some of the larger manufacturers. Such machines are not suitable for all components, nor can they be justified economically except for very large outputs since they lack flexibility and cannot easily be adapted to meet changes in component design.

Even for smaller outputs it is, however, possible to simplify and cheapen the passage of work through a machining sequence. For example, tunnel type work fixtures can often be adopted in such a manner that the work fixture is in line with the roller conveyor between successive machines. With this arrangement the work travels in a straight line and loading and unloading times are reduced. A further refinement that is possible when a tunnel type work fixture is employed is what may be termed the single-station transfer machine. For instance, a travelling-head milling machine can have an ejector incorporated in the work head so that the component is automatically released and moved clear of the fixture during the return stroke of the machine.

In a slightly different field, attention may be directed to developments made by Vauxhall Motors Ltd., to minimize handling during several successive press operations. Short endless belt conveyors are installed between the presses so that the work is conveyed directly from the unloading side of one press to the loading station of the next. In addition, where practicable, ejection arms have been incorporated in the press mechanism. These are additional to the normal vertical ejectors which merely push the work out of the bottom die. When the normal ejectors have pushed the

work clear of the die, the arms operate to pull the pressing on to the moving belt for transfer to the next machine.

Available equipment

So far as actual material handling equipment is concerned there is an embarrassment of riches from which production engineers can choose that best suited to their needs. That this is so will be amply demonstrated at the Mechanical Handling Exhibition to be held at Olympia from June 4th to 14th. There will be every opportunity of comparing almost all types of mechanical handling aids manufactured in this country. Technical assistance and advice on problems connected with the movement of materials will be freely available, but we must repeat that optimum results can be achieved only if mechanical handling as such is considered as an integral part of materials handling and not as a completely separate function.

The Labour Force

FOR a number of years a disquieting change in Britain's population has been a marked and sustained increase in the proportion of older to younger people. Attention was directed to the problem by the study and application of National Insurance policy and, more recently, economic investigations and productivity surveys have revealed and reaffirmed the critical character of the problem.

In 1911, 2½ million people were over the commonly accepted age of retirement; 65 for men and 60 for women. By 1947 the figure had reached nearly 6½ million and the estimated total for 1977 is 9½ million. Restated approximately, these figures are equal to one, two and three persons respectively in each fifteen of the entire population or for each ten people of working age.

As would be expected, the trend applies to age groups below that for retirement. By 1960 it is expected that the population between the ages of 20 and 40 years will have decreased by seven per cent, while the number between 50 and 60 years will have increased by about seventeen per cent. There is a higher ratio of employment in the lower age group, so the working population will be markedly affected. This applies with particular emphasis to women.

The trend of movement will, unfortunately, exert a double influence on our economy. Not only will a reduced number of workers have to maintain and increase production, but at the same time must carry an increased number of unproductive consumers. Inevitably, the effect will be to depress the general standard of living to a level lower than otherwise it need be.

It is, of course, a national problem and is recognized as such by the Ministry of Labour and National Service. As the expectation of life has increased, it would seem our views on the expectation of working life should be modified. Automatic retirement at the 65—60 age limit should no longer be regarded as normal. The Minister has set up a National Joint Advisory Council, representing the British Employers' Confederation, the Trades Union Congress, and the nationalized industries to consider the problem in detail. This Council has endorsed a national policy that urges (a) " older persons should be retained in employment for as long as they want to continue, provided they are fit for their normal work or for any alternative work which can be provided for them " and (b) " there should be no impediment to the recruitment into employment of older persons who are both able and willing to carry out the jobs available ".

Future policy

The immediate co-operation of both employers and employed is sought. It is emphasized that any steps taken should be on the basis of voluntary contract. Proposed lines of action include the review and revision of schemes of compulsory retirement at fixed ages, of pension schemes requiring retirement at fixed ages, and of practices or arrangements making it difficult or impossible for older persons to be engaged. Special working arrangements, hours and conditions should be introduced, where necessary to suit the needs of elderly persons and enable them to continue working, and the practicability of segregating sections of work for older persons should be investigated, it is suggested.

Each industry and each individual firm should bring its personnel policy and practice into line with national needs. They must ensure that those responsible for engaging and retiring workers and staff should understand the economic and social implications of the problem and apply the policy.

It would seem that some of the energy and resources hitherto devoted to apprenticeship schemes may have to be diverted to the rehabilitation of the elderly worker approaching, or recalled from, retirement.

CONTENTS

FORD-MERCURY TRANSMISSION

A Simple, Three-element, Torque-converter Unit Embodying Ingenious Control Devices

By O. D. North, M.I.Mech.E.

In our issues for March and April, 1951, American torque-converter transmissions were dealt with in a general way. Space did not permit bringing out to the full the individuality of the various approaches to a most complicated problem. This more detailed description of the aims and methods of the designers of the Ford-Mercury transmission will, therefore, be of interest. While reference may be made to the two articles referred to, the general torque-converter background is briefly covered in the introductory remarks.

IT is the aim of American manufacturers to produce a vehicle which can, under normal open road and city traffic conditions, be driven all day without removing the hand from the steering wheel except for parking and without using the feet except for the accelerator and the brake. Other approximate requirements include the ability to climb grades of about 10 per cent at about 60 m.p.h. without any shift of gearbox ratio, either manual or automatic, and an acceleration from rest—not necessarily without automatic change of ratio—at least equal to that given by a full-throttle start on second speed of the conventional, three-speed, synchromesh gearbox.

Where, as in the case of Buick, Chevrolet and Packard, the torque converter is the only ratio-change available in the normal or drive range a particularly good power-weight ratio is essential, involving considerable engine piston displacement per mile. A reasonably high conversion efficiency in the neighbourhood of the "stall" point is also clearly desirable, but this, even with multi-stage converters having two stators and two pumps, is difficult to obtain without prejudicing somewhat the performance of the converter as a fluid coupling at the higher speeds.

When one considers that a fluid coupling is, in effect, a device to dissipate as much energy as possible for a given slip figure it is really remarkable that it has been possible to make it serve, at lower speeds, as a device to waste as little energy as possible when turning a high rate of "slip" into useful torque multiplication. There is, nevertheless, a somewhat unfortunate region in the neighbourhood of 1,800 engine revolutions per minute on full throttle, in which the converter has, so to speak, not made up its mind which profession to adopt, and wastes over 10 per cent in slip without doing any torque conversion at all.

Packard uses a torque converter with the very high stall-ratio of 2·4 : 1, and that without employing double stators, and counteracts the effect of its inevitable inefficiency as a fluid coupling by providing an automatic locking clutch which is normally engaged for the greater part of the mileage run by the vehicle.

Studebaker use a converter with a stall-ratio of only 2·15 : 1, of the simplest possible type, with a locking clutch to eliminate coupling slip. Axle ratio is fairly high in this case at 3·54 : 1 on a car weighing 3,500 lb with an engine giving only 102 h.p. at 3,200 r.p.m. On this car, however, as soon as the locking clutch is automatically disengaged, the power passes through the torque converter and an epicyclic reduction of 1·435 : 1. Thus the torque converter on the Studebaker never has to work on direct drive, and for normal acceleration and hill-climbing its duty is immensely relieved by the additional reduction. It has, in all probability, seldom to hang on under full throttle at a speed corresponding to the awkward change from coupling to conversion, and probably does most of the hard work acting as a pure coupling at relatively high engine speeds.

The Ford engineers seem to have evolved a logical compromise between the advantages and disadvantages outlined. They realize that a big engine displacement per mile involves high fuel consumption and engine friction losses. The axle ratio chosen is, therefore, distinctly on the high side at 3·31 : 1. The torque converter is of the simple type but, by virtue of it not being designed to give a high stall ratio, it can be made more efficient as a fluid coupling. It has a road load slip of only 2·5 per cent at 60 m.p.h. and a locking clutch is, therefore, not really required and is omitted.

Being thus always in circuit, the torque converter is available to give increased torque without the perceptible change in engine revolutions associated with the automatic disengagement of a locking clutch. At the same time the converter is not called on, in normal driving conditions, to work at more than about 0·5 speed ratio, say 1·55 torque ratio and 77 per cent efficiency. If loaded beyond this point on the drive range, an epicyclic reduction comes into operation, giving a demultiplication of 1·48 : 1. Not the least advantage of this provision of a gear reduction in the drive range is the opportunity it affords of increased hill-climbing speed by increased engine revolutions and power. This applies also in the case of the Studebaker.

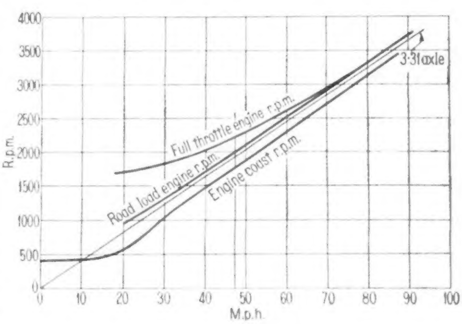


Fig. 1. Converter slip

As mentioned on page 84 of the March, 1951, issue of the *Automobile Engineer*, one of the chief defects of the unassisted torque converter is that every conversion ratio, on full throttle, is associated with a definite and limited engine speed. By putting a gearbox, on drive range, between the converter and the back axle, engine speed and power available may, at times, be increased by at least 30 per cent over that obtaining on a car without this provision.

In low range, manual operation of the selector lever on the Ford trans-

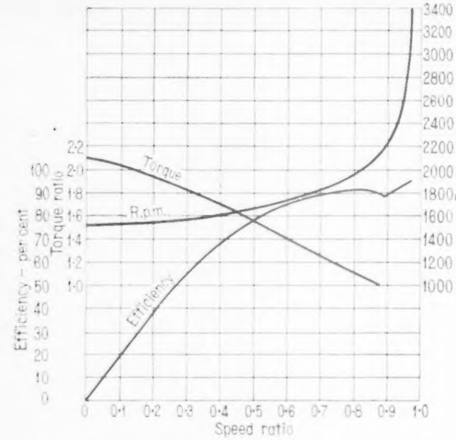


Fig. 2. Torque-converter characteristics at full throttle

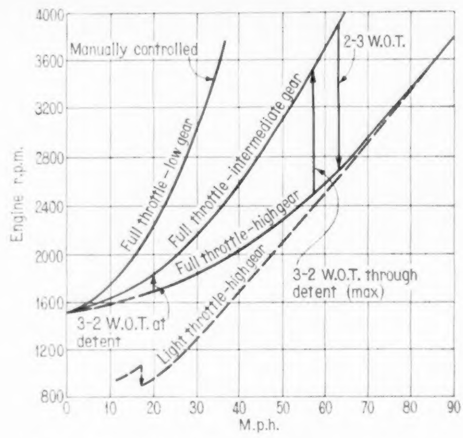


Fig. 3. Shift diagram: W.O.T. signifies wide open throttle

mission gives an epicyclic reduction of 2·44 : 1. This, deliberately, is on the low side in order to assist engine braking, the converter having no special provision, as on the Chevrolet, to increase its coupling power on the overrun. Reverse comes conveniently at 2·00 : 1.

In a paper by Mr. H. G. English, assistant research engineer, Ford Motor Company, several curves which help greatly to explain the behaviour of the Ford transmission are given.

Fig. 1 shows the relation between engine speed and road speed with the converter only in action, the epicyclic train of the 1·48 : 1 intermediate gear being locked by its friction clutch. Road load, a term not, perhaps, familiar to English engineers, signifies the torque and power conditions necessary to maintain the indicated steady speed on a level road. Rolling resistance and windage are allowed for, but no hill-climbing or acceleration. The road load curve becomes tangent to the full throttle curve at the normal maximum speed of the car.

Under road load the converter acts as a coupling down to 20 m.p.h.; on full throttle the coupling effect persists down to 47 m.p.h., at which point the converter slip is 10 per cent (the thin line represents output shaft revolutions). Below 47 m.p.h. the converter begins to multiply the torque and Fig. 2 gives the multiplication ratios in terms of engine speed, which can be taken from Fig. 1.

It must be distinctly understood that the curves in Fig. 2 refer to full throttle operation only; on part throttle the change-over from coupling to conversion is delayed to much lower speeds while at light throttle it seldom occurs at all.

Fig. 3 requires a good deal of explanation. Low gear means the lowest ratio in the epicyclic, 2·44 : 1, intermediate gear, 1·48 : 1, is designated No. 2, and high gear, direct drive through converter, as No. 3. When starting on drive range the manual control mechanism engages intermediate gear. If the accelerator is held down—but not trod past the detent stop—the engine speed will rise to about 3,900 r.p.m. whereupon the automatic mechanism takes charge (2-3 W.O.T. signifies intermediate to high, wide open throttle) and shifts the transmission into direct drive.

It is not in the least necessary to drive in this furious fashion; it the throttle is opened a moderate amount only the change up is automatically made at much lower speeds. The dotted line "light throttle-high gear" shows the extreme case, in which the up-change takes place at about 17 m.p.h.

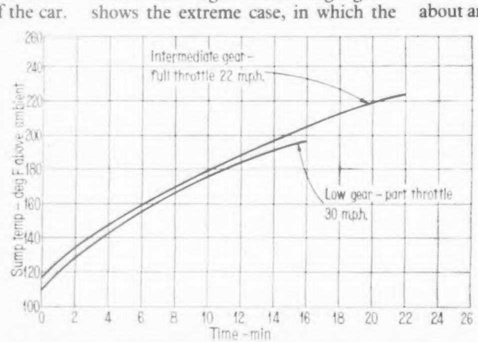


Fig. 4. Transmission cooling : conditions equivalent to 6 per cent grade and 4,700 lb. towed load

Assuming the car to be proceeding on high gear at any speed below 57 m.p.h., a change down to intermediate can be forced by pressing the accelerator right down past the spring detent (3-2 W.O.T. through Detent, Max). If, however, the car slows down without the pedal being pressed past the detent a speed of 20 m.p.h. can be reached, at which the mechanism automatically changes from high gear to intermediate.

An interesting example of the value of the epicyclic reduction, as compared with operation on the converter only, can roughly be extracted from these curves. The Ford engine peaks at 3,600 r.p.m., giving 100 h.p. Assume the car to be climbing a hill of about 13 per cent grade which would hold it, on intermediate gear, to 3,600 engine r.p.m. This, from Fig. 3, corresponds to about 58 m.p.h. Coupling slip at this engine speed would be about 3 per cent and gear losses about another 3 per cent, nett efficiency 94 per cent and nett horsepower to back wheels about 94.

Now assume the car to be put at the same hill on high gear, leaving the torque converter to do all the multiplication. Clearly, it will be operating on a point on the engine power curve where the torque is greater, and since the windage at a speed probably in the region of 35 m.p.h. will be negligible, it may be assumed that a multiplication of only 1·2 :1 in the torque converter will suffice,

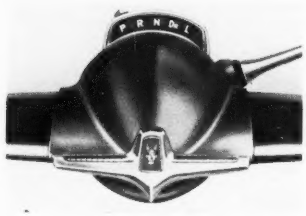


Fig. 5. Gear selector on steering wheel hub

against the 1·44 : 1 in the first case.

From Fig. 2 it may be noted that a torque ratio of 1·2 : 1 corresponds to an engine speed of 1,900 r.p.m. and from Fig. 1 that an engine speed of 1,900 r.p.m. on full throttle corresponds to a car speed of about 35 m.p.h. only. To express it crudely, the epicyclic gear beats the torque converter by 23 miles an hour. The reasons for this are two: in the first place the torque converter inevitably assigns an engine speed of 1,900 r.p.m., at which speed the engine probably gives about 60 h.p. only, while in the second place the torque converter is working at about 90 per cent efficiency as against 94 per cent.

It is interesting to note that the unfavourable result is not, in this case, primarily due to low efficiency of the torque converter: it results from the limitation of speed set by the torque-converter performance curve. Had a somewhat steeper hill been chosen, holding the vehicle down to lower speeds in both cases, there would have been less windage in the first case and the torque converter would have been working at a torque ratio of say 1·4 : 1, at which the efficiency would have been about 84 per cent only. The relative performance of the torque converter would have been somewhat worse.

While these calculations must not, on any account, be taken as anything but approximate, they give a clear indication of the value of incorporating a mechanical reduction in the automatic operation of the drive range: the intermediate gear of the Ford would, for example, be a godsend when towing one of the typical American "House-trailers", weighing over two tons, in a hilly district.

Fig. 4 shows the cooling curve for the transmission oil on a test corresponding to the towing of such a trailer up a 6 per cent grade. Relatively better cooling might have been expected in the case of the low gear, since the converter would be working as a lightly loaded coupling and the fan blades on the converter body would have been revolving at a far higher speed.

Before proceeding to a description of constructional details one or two points of general interest may be mentioned. In the first place it should be noted that Ford use a sequence of gear-selector positions, Fig. 5, different from the rest of the American industry. The parking position, in which the transmission is locked by a toggle-operated pawl, Fig. 6, engaging teeth cut in the epicyclic gear annulus,

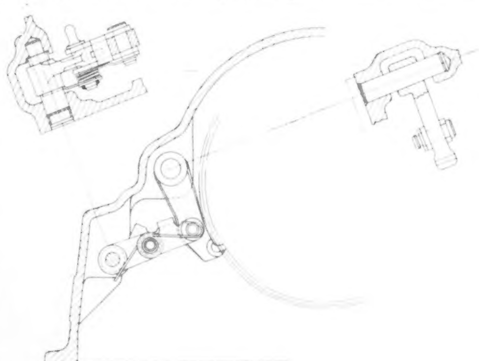


Fig. 6. Arrangement of parking lock

comes on the left, followed by Reverse, Neutral, Drive, and Low. One of the reasons for this change is that by placing neutral between drive and reverse, accidental engagement of reverse gear is less likely and a reverse inhibitor, as used on some other makes, is not found necessary. Another advantage claimed is that it is not necessary to go through a forward gear to engage reverse. Rocking between low and reverse, as insisted on in America for extricating a vehicle from snow, is claimed not ot be prejudiced by this arrangement.

Torque converter

As in all torque-converter transmissions the body of the converter, Fig. 7, is flexibly attached to the engine. crankshaft. The stout pressed steel converter cover has the starter gear shrunk on it and has welded to its centre a small pressing having its forward end machined spherically to fit, as a centring spigot, in the bore of the crankshaft. Drive is transmitted by a thin steel pressing about six inches wide, bolted to the crankshaft flange and at its ends picking up six of the thirty-four $\frac{5}{16}$ in bolts which secure the converter pump body to the cover.

Die-cast in aluminium alloy the pump body, Fig. 9, carries thirty one steel blades, each one having four ears fitting in die-cast recesses. A wire ring, sprung into an outer recess, keeps the outer ends of the blades in position, while a pressed steel torus ring fits on to two tabs on each blade, these being rolled over after assembly. The pump body has numbers of radial cooling ribs on its outer surface and is completed by a cast-iron hub, attached by setscrews and sealed by a rectangular section synthetic rubber ring, having a rearwardly extending sleeve. The outer surface of the sleeve takes the oil seal in the transmission case while its end is formed with two tongues to drive the front oil pump.

The turbine assembly, Fig. 10, is based on a steel stamping, carrying thirty-three blades, each with four tabs passing through pierced slots and rolled over. A torus ring, similar to that on the pump member, completes the blade assembly. The steel hub, secured by rivets, is broached to take the splined end of the transmission input shaft.

The converter stator, Fig. 11, is a die-casting, with thickened and rounded edges on the entry side and with a formed steel shroud ring sprung over the tips of the blades and joined by welding. To the broached bore of the

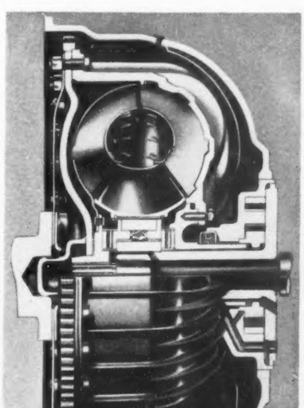


Fig. 7. Air-cooled, three-element torque converter

TRANSMISSION AND OVERALL RATIOS

	Transmission Gear Ratio	Stall Ratio Converter	Transmission Overall-stall	Overall 3·31 Axle
Low	2·44	2·10	5·13	16·98 to 8·08
Intermediate	1·48	2·10	3·11	10·29 to 4·90
High	1·00	1·57 (at 20 m.p.h.)	2·10	5·20 to 3·31 (at 20 m.p.h.)
Reverse	2·00	2·10	4·20	13·90 to 6·62

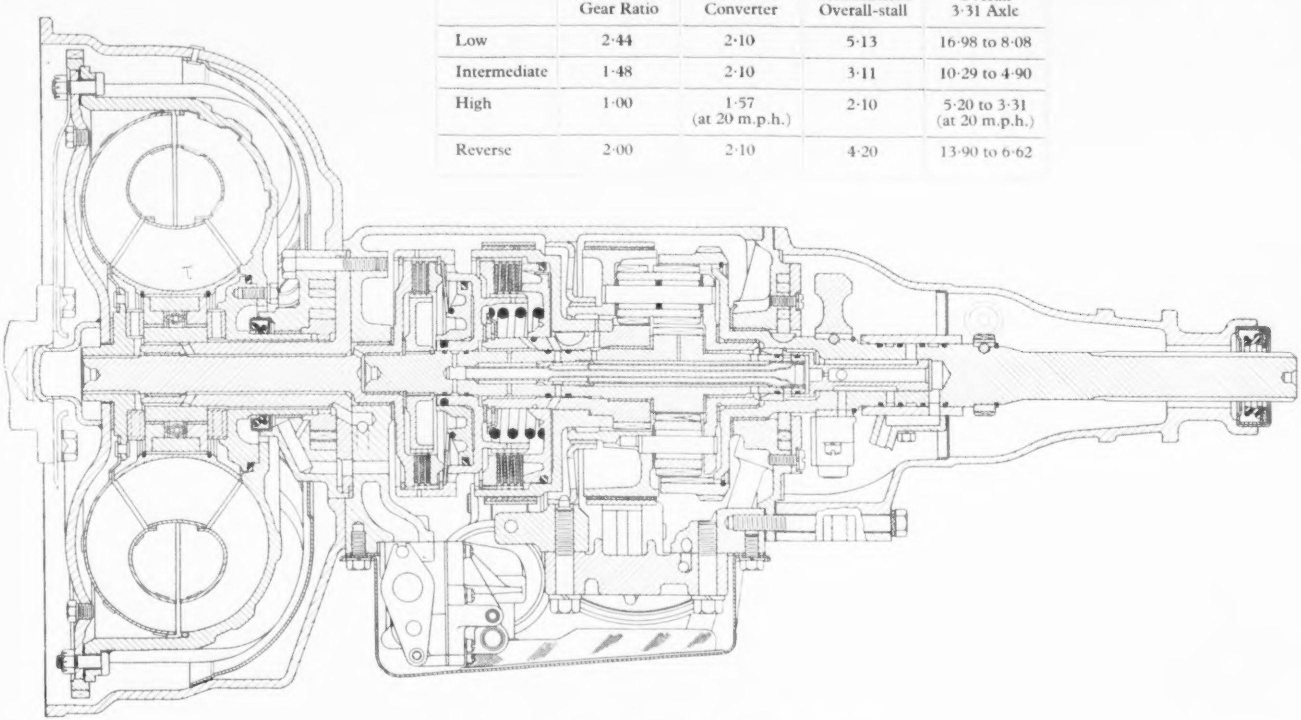


Fig. 8. GENERAL ARRANGEMENT OF TRANSMISSION UNIT

stator is fitted the splined outer race of the sprag-type free-wheel, two snap rings giving end location. Lightly pressed in to the outer race are two shouldered rings lined with white-metal which take their running bearing on the inner race of the free-wheel, which is broached to fit on the cast-iron stator support attached to the transmission case.

The converter assembly is completed by three bronze thrust washers, the two of these coming on each side of the free-wheel being fairly thick and having radial holes through which takes place the circulation of fluid induced by the transmission oil pumps.

The whole converter is enclosed in a housing, Fig. 13, attached to the engine crankcase and having a large opening at the rear, by which air enters, flowing to the centre and being flung out by the radial vanes on the converter pump body to emerge by another rearward opening on the right side of the vehicle. Internal diaphragms of pressed steel guide the air flow in the housing. This constitutes the only cooling, there being no external pipework and oil coolers, as on most of the other torque converters.

A detachable cover plate gives access to the lower part of the front of the converter assembly, permitting the bolts securing the ends of the flexible driving plate to be removed, preparatory to removing the converter from the vehicle. It is possible, however, to remove the transmission without disturbing the converter or its housing, see Fig. 8. The input shaft, stator support and pump drive lugs slide out of their companion members as the transmission is withdrawn to the rear. The operation requires some care when replacing, since two sets of splines and two dogs have to be engaged almost simultaneously.

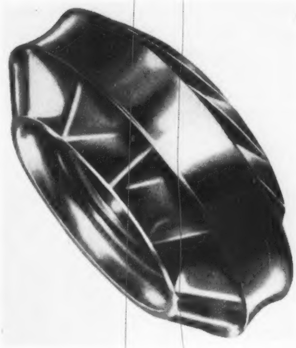


Fig. 11. Stator element

Transmission

The input shaft from the torque converter has, at its inner end, an integral flange with external teeth fitting into the internal teeth cut in the body of the front clutch, a snap ring securing the two parts together to form a unit. Made of cast-iron the front clutch body comprises the annular bore for the front clutch piston and a steady bearing on the primary sun gear shaft, through which the application fluid is supplied, sealing being, as in all other cases, by piston ring seals on each side of the feed groove.

The front clutch, which locks the input shaft to the primary sun gear shaft, has three sintered bronze and two steel plates, applied by a slotted Belleville washer which acts as a multiplying lever and also as a powerful withdrawal spring. The friction surface provided is not large, but it should be observed that this clutch is seldom engaged under torque and therefore demands little thermal capacity. Full stall torque of 2·1 times engine torque can be handled, owing to the leverage multiplication, with less than 100 lb/in^2 oil pressure.

Teeth cut on a rearward extension of the front clutch body take the driving plates of the rear clutch, the driven plates fitting in a housing keyed to the secondary sun gear and forming also the drum with which the front gear band brake engages.

There are four pairs of plates and the operating piston is larger than that of the front clutch; no multiplication is therefore necessary. It should be observed that the helical withdrawal spring is very stiff, as in the case of the front clutch. A stiff withdrawal spring has made unnecessary the automatic venting valves provided by other makers to eliminate clutch drag due to build up of centrifugal pressure in the clutch cylinders.

Epicyclic gearing

The primary sun gear, which is the smaller of the two, comes behind the secondary gear and meshes with narrow primary idler pinions; these meshing, in turn, with long secondary pinions carrying over to the secondary sun gear. Output is taken from a steel internal gear meshing with the secondary pinions and secured to a toothed flange integral with the output shaft by a shoulder and a snap ring. The parking lock teeth are cut on the outside of the internal gear ring.

All teeth have 14 diametral pitch and a helix angle of 20 deg. They are made of high-carbon alloy steel, through-hardened and crown-shaved before hardening—the long pinions are shaved in two operations to suit the meshing positions of the two sets of gears with which they engage.

The pinions run on needle roller bearings, two sets with a spacer ring being used in the long pinions. The pins are induction hardened, with the ends left soft for staking into the pinion carrier.

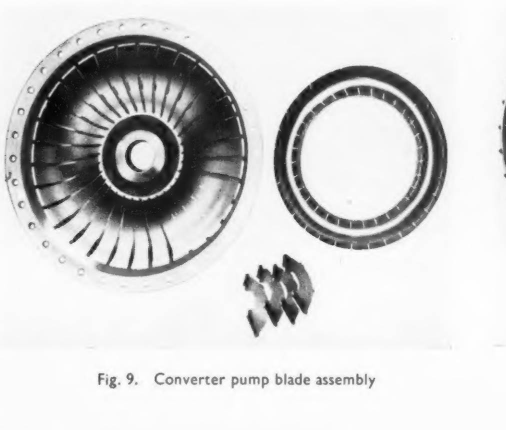


Fig. 9. Converter pump blade assembly

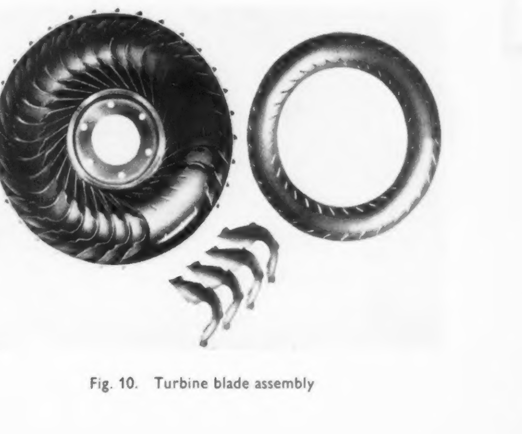


Fig. 10. Turbine blade assembly

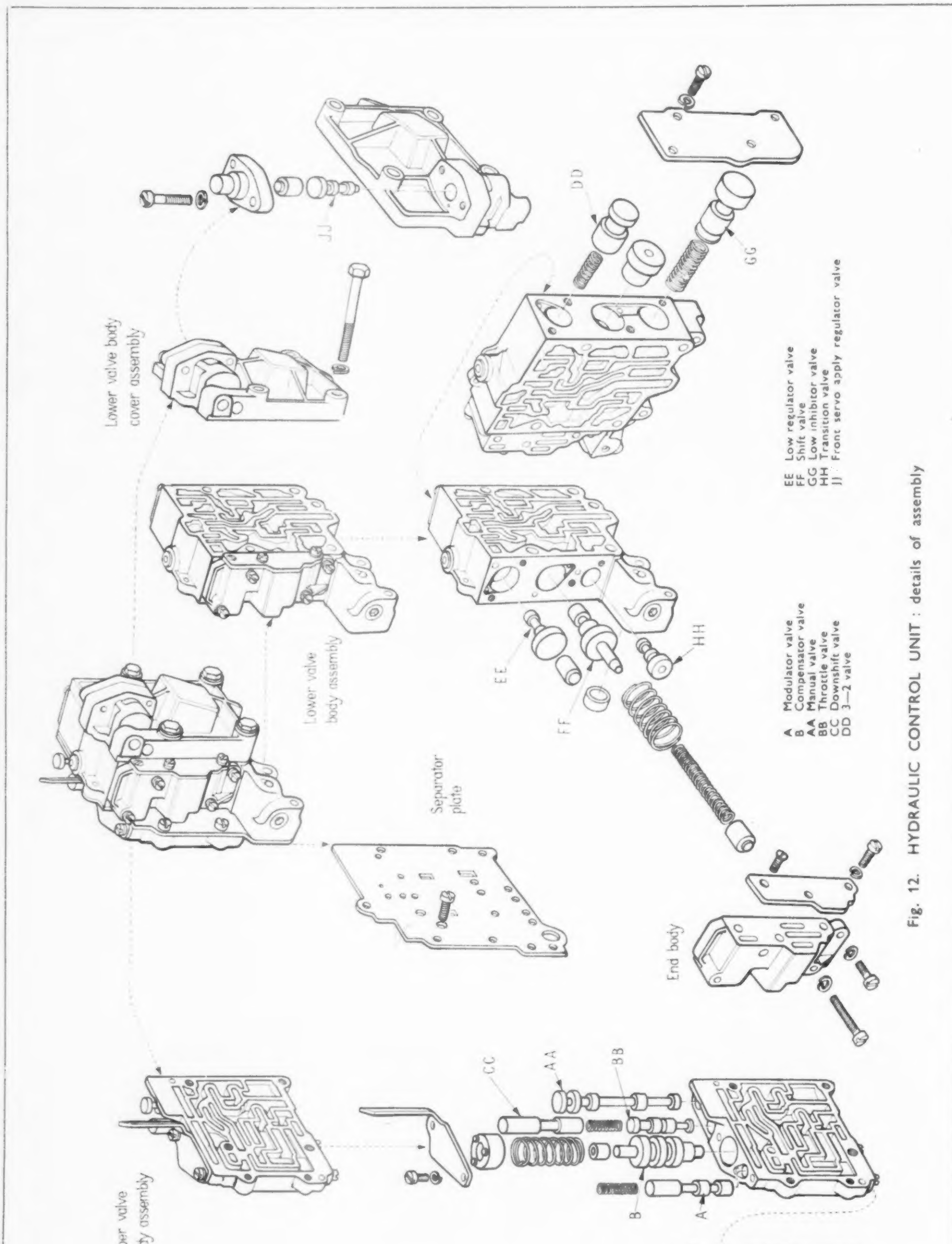


Fig. 12. HYDRAULIC CONTROL UNIT : details of assembly

Fig. 13. Cooling air circuit

Cast-iron is used for the pinion carrier, which is made in two pieces, riveted together before final line-boring of the pin holes. The drum for the low and reverse band is integral with the pinion carrier, which also includes a spigot bearing running in a large diameter but short bushing carried by a pressed steel diaphragm lightly pressed into the middle of the transmission case. This support bearing removes from the running shafts the relatively enormous reactions from the low and reverse gear band which, unlike the system adopted in the well-known Wilson gearbox, has a single anchorage point only.

Considerable end thrust is at times exerted on some of the running members by the helical angle of the teeth. This is taken care of by numerous bronze thrust washers, having radial grooves to assist lubrication.

Since copious lubrication is inevitably available from the control system there has been no need to fit ball or roller bearings at any point except the planet pinions, a fact which makes possible a much more compact and cheap assembly than would otherwise have been possible.

In Fig. 14, diagrams show the power path through the transmission in the various selected gear ranges. Students of epicyclic gear design should take particular note of the way in which the central support bushing previously mentioned relieves the running shafts of bearing loads otherwise inevitable from the reactions of both brake-band anchorages.

Both gear bands have bonded linings; that for the rear band is 0·050 in thick, of semi-metallic material. The front band has to take only about one engine torque at converter stall point, its servo, including lever, can therefore be made as aluminium die-castings throughout. This servo, Fig. 16, is remarkable for having a double-acting cylinder, the release side having exactly double the area of the apply side—the importance of this is brought out in the description of the control mechanism.

The rear servo, Fig. 17, is single-acting with a large cylinder and a stamped steel multiplying lever. Its band has to take about six times engine torque in reverse, so robust construction is essential.

While converter housing and main transmission casing are of cast-iron, the rear extension, until defence requirements intervened, was of aluminium alloy. The converter body and no less than nineteen smaller die-castings in the transmission valving and control mechanism are also of light alloy. The complete assembly then weighed 75 lb more than the standard Ford three-speed synchromesh transmission.

Hydraulic control system

The excellent booklet issued by the Ford Motor Co., entitled "Operating principles of the automatic transmission" devotes eleven pages of small print and thirty diagrams to explaining the hydraulic control system without going into constructional details. The drawing of the control unit, Fig. 12, is of interest as it reveals how, by a sandwich construction, cross-connections are established without the need for pipework.

About one and a half English gallons of special transmission fluid are required to fill the system, most of this being in the converter body and the remainder in the pressed steel sump attached to the transmission casing. A change of fluid is recommended after 15,000 miles' running.

Two pumps, of the internal gear type, with crescent abutments, draw from the sump through gauze strainers and deliver through check valves to the control pressure regulating valve. The front pump is driven by two tongues on a rearwardly-extending sleeve attached to the converter body; its gears are $\frac{5}{8}$ in wide and run in a cast-iron casing. Those in the rear pump are of identical diameter but only $\frac{5}{16}$ in wide; they run in a die-cast aluminium casing closed by a steel cover plate, which is phosphate-coated to prevent scoring.

The control pressure regulating valve can be described as a piston relief valve, limiting the control pressure by virtue of a coil spring. Neglecting, for the moment, other forces acting on the piston of the valve, a pressure of about 150 lb/in^2 is established by the spring pressure alone. Movement of the piston allows excess oil to escape to the torque converter body. Return of the oil from the converter to the sump is controlled by the converter pressure regulator valve, which may also be described as a piston relief valve under spring load with additional counter-loading to be referred to later.

Among the functions performed by the porting in the control pressure

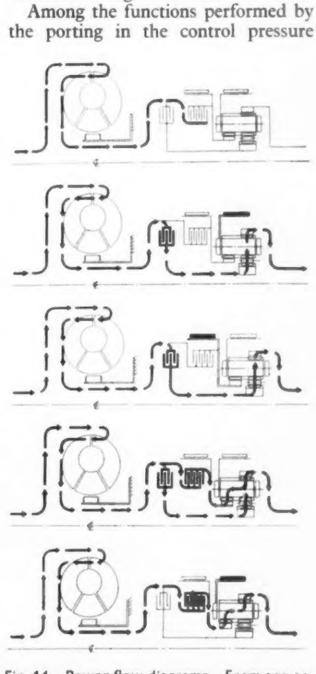


Fig. 14. Power flow diagrams. From top to bottom : neutral, low gear, intermediate gear, high gear and reverse

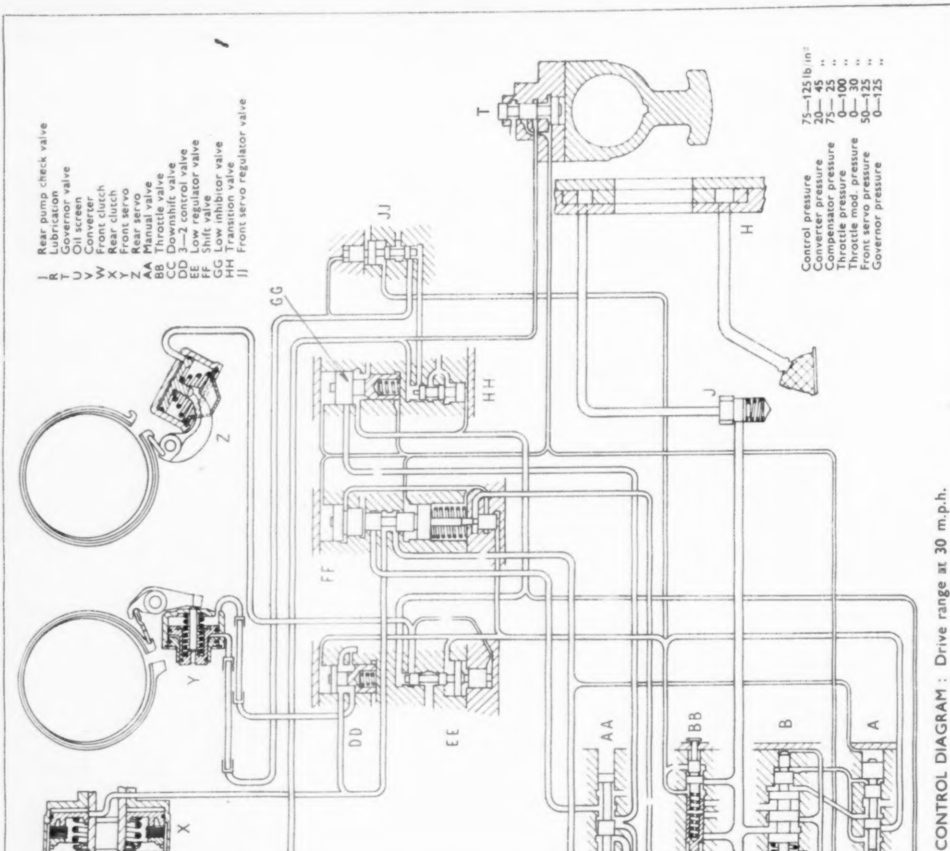


Fig. 15. CONTROL DIAGRAM : Drive range at 30 m.p.h.

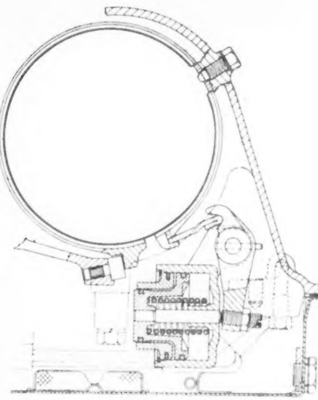


Fig. 16. Front band servo

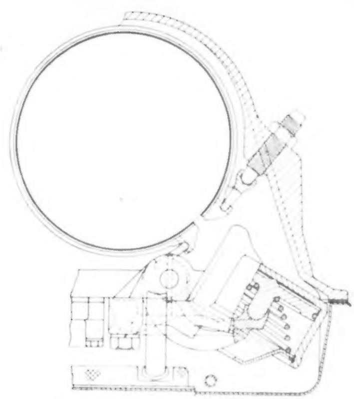


Fig. 17. Rear band servo

regulating valve is that of by-passing the delivery of the large front oil pump when the vehicle speed is sufficient for the rear pump to take over. The front pump is then relieved of the full control pressure of 75 to 150 lb/in² and has only to deal with the supply to the converter which varies from 20 to 45 lb/in².

An important function of the converter pressure regulating valve is that of closing the return to the sump from the converter as the vehicle comes to rest and the engine is stopped; the converter is thus sealed and ingress of air prevented while the car is standing.

Quite apart from the automatic shifting of the gears in the transmission, it is found desirable to vary the control pressure according to the pressures required to hold in engagement the clutches and band brakes, which vary with throttle position and vehicle speed. Avoidance of power wastage by excessive demands on the oil pumps is one consideration, equally important is the avoidance of forceful engagement of clutches and brakes when the torque to be transmitted does not demand this.

A valve, called the throttle valve, is connected to the accelerator in such a way as to increase the oil pressure passed by it as the engine throttle is opened. This throttle pressure is passed through a modulator valve operated on by oil under control of the manual valve which selects drive range, low range or reverse. The effect is that in drive range the throttle pressure is reduced somewhat by a spring, while in low and reverse the throttle pressure passes through the valve at its full value, appropriate to the much greater pressures required to hold the low and reverse gear band under the abnormally high torque.

Throttle pressure, thus modulated, passes to one end of the compensator valve, where it is opposed by governor pressure developed, in proportion to the square of the vehicle speed, by a centrifugal governor, Fig. 19, on the transmission output shaft. This may be described as a piston reducing valve whose regulating force is the centrifugal force acting on the piston mass, which slides radially in a die casting bolted to a stamping surrounding the output shaft of the transmission and having a counterweight integral with it to balance the mass of the governor valve assembly.

By suitable porting the compensator valve strikes a balance between throttle and governor pressures and supplies oil at compensator pressure to a land on the control pressure regulator valve which acts in opposition to the spring, thus reducing the control pressure, by varying amounts, below the 150 lb/in² which would otherwise obtain.

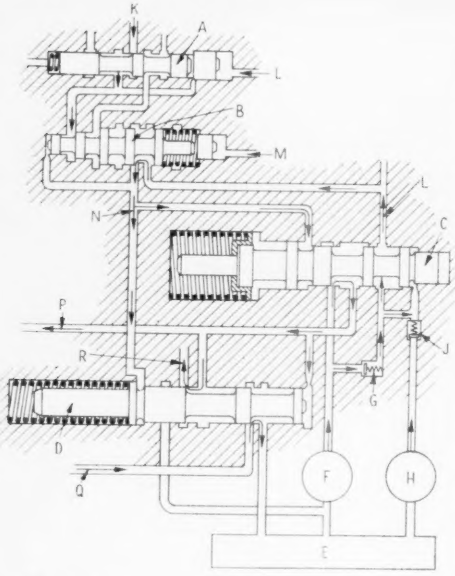
The drive range

When the selector lever is moved into the position marked Dr, as in Fig. 15, the manual valve is moved so as to admit controlled pressure to the front clutch, by way of the rear groove on the floating banjo on the output shaft. Pressure is also supplied to the apply side of the front band servo. The converter is thus connected to the small sun wheel of the epicyclic gear, while the large sun wheel is held stationary, giving intermediate gear, with a reduction between converter and output shaft of 1·48 : 1.

Pressure is at the same time connected to the line feeding the rear clutch, but the passage is stopped by the shift valve which is held in position by a spring. One end of this shift valve is connected to throttle pressure and the other end to governor pressure. As the car speeds up on intermediate gear, the governor pressure builds up until it overcomes the spring and the throttle pressure.

The shift valve then moves, admitting pressure to the rear clutch and to the release side of the front band servo piston. This has double the area of the apply side so that the band is forcibly released in spite of the fact that the apply pressure is still on. The arrangement is more subtle than might at first appear, since it permits the clutch being engaged before the band is released but does not allow more than about 35 per cent of the maximum grip to be applied to the two simultaneously. Drive to the back wheels, therefore, need never be interrupted during an automatic change into high gear, yet there is no possibility of a momentary "lock-up" of the transmission.

It will be noted, since throttle pressure is a factor in controlling the shift valve, that a change up will take place much earlier on light throttle. If the accelerator is held hard down, the car stays in intermediate gear up to about 62 m.p.h. In spite of the interaction between high gear clutch and intermediate gear band above described, it is found that a little roughness due to simultaneous engagement can be detected when changing on light throttle. A valve, termed the front servo apply regulator valve, is therefore introduced which reduces, according to throttle pressure, the apply pressure existing in the front band servo at the time of the upchange. Release of the band thus occurs a little earlier than would otherwise be the case. On full throttle this reduction does not take place, since it would result in momentary engine racing.



A Modulator valve
B Compensator valve
C Control pressure regulator valve
D Converter pressure regulator valve
E Sump
F Front pump
G Front pump check valve
H Rear pump
J Rear pump check valve
K Throttle pressure
L Control pressure
M Governor pressure
N Compensator pressure
P To converter
Q From converter
R Lubrication

Fig. 18. Compensator pressure diagram

Yet another refinement is applied to the change from high to intermediate, designed to prevent a too rapid engagement of the intermediate gear band when the transmission automatically shifts to intermediate at low speeds and light throttle opening. This, if not prevented, would cause a slight but disconcerting overrun snatch. A valve, called the 3-2 valve, obstructs the release fluid passage from the front band servo on light throttle openings, being operated by the throttle pressure. On full throttle, however, this valve leaves the psasage wide open to prevent engine racing on the change-down.

In normal driving, with moderate throttle openings, the car changes into high gear fairly early and stays there at reasonably high speeds, even with full throttle. Should, however, the driver wish to change down at, say, 50 m.p.h. he can do so by pressing the accelerator past a spring-loaded detent. This operates the downshift valve which, by directing control pressure to the spring side of the shift valve, overcomes the relatively high governor pressure. This will operate up to 58 m.p.h., while at 63 m.p.h. the transmission, if still running in intermediate, is forcibly changed to high no matter what the accelerator position.

The Low range

With the selector placed in the L position, the manual valve admits control pressure to the front clutch and the rear band servo, giving a reduction of 2·44 : 1 between converter and output shaft.

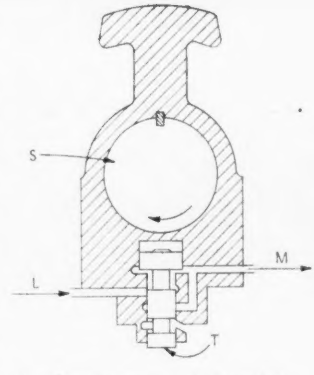
It will be remembered that it was found convenient to use the same passage to take fluid from the manual control valve to the front clutch and the front servo. Unless additional precaution was taken this would mean both band servos being in operation at once, leading to dangerous "lock-up" strains in the epicyclic gearing. Another valve, called the transition valve, takes care of this contingency.

Actuated by the rear servo apply pressure, the transition valve closes the line to the front servo, preventing front band application. To further prevent ratio change in low, control pressure is not directed to the shift valve. Once the transmission is in low range, the ratio will not change unless the selector is shifted to another position.

Yet another valve is the low regulator valve which regulates the control pressure applied to the rear band servo according to throttle opening. This prevents the violent overrun snatch which would otherwise occur if the selector were moved to L at, say, 25 m.p.h. with shut throttle.

When the selector lever is moved to the reverse position, the manual control valve directs control pressure through the low regulator valve, which regulates the pressure to the rear servo applying the rear band, in the same manner as for low range. At the same time control pressure passes through the shift valve to the rear clutch and to the release side of the front servo.

The last valve to be mentioned is called the low inhibitor valve. It prevents the transmission from shifting to low range when the selector lever is moved to L at road speeds above 27 m.p.h. The valve is actuated by spring and governor pressure. At speeds above 27 m.p.h. the governor pressure closes the low inhibitor valve, blocking the rear servo apply line. With no pressure in the rear band line, the transition valve will not shut off the front band apply pressure. The result is intermediate ratio instead of low.



L Control pressure
M Governor pressure
S Output shaft
T Governor valve

Fig. 19. Governor valve

Metal Films on Glass

WHEN in dry weather the glass windows of instruments are cleaned, they may accumulate a charge of static electricity that can lead to false readings. This problem was brought to the National Physical Laboratory and, in the subsequent research into means of avoidance, a method of coating glass with a thin, transparent film which will conduct electricity has been developed.

The layer on the glass is an extremely thin one of an oxide of metal. Not all metals produce transparent oxide films, but one which has given good results is tin. It can be deposited very evenly on the glass by techniques which are already well understood in industry. The glass is then heated to near the softening point and cooled again. The tin oxidizes and becomes transparent. Finally the film is washed in water and dried, a step which increases its conductivity. After this it is hard, inseparable from the glass surface, and is resistant to chemical attack.

Visibility through the glass is reduced by the metal coating by an amount which is negligible for all practical purposes.

Enough current can be passed through it to keep the surface of the glass so hot that it is impossible for ice and snow to form on it or for condensation to make it misty. In practice the film would be sandwiched between two layers of glass. The process is being patented. (2003)

MECHANICAL HANDLING

An Important Exhibition at Olympia

FROM June 4th to 14th the third Mechanical Handling Exhibition will be held at Olympia. This is an event that production engineers, and more particularly those concerned with the handling of materials, should not miss. Some 170 firms will be exhibiting many different types of equipment. The exhibits will include conveyors, elevators, hoists, stackers, fork lift trucks, industrial trucks and ancillary equipment.

Efficient handling of materials, even within a single organization, usually entails the employment of mechanical handling equipment of many different types. Frequently, it is a matter of great importance that the correct type be chosen, but this is not always easy. At this exhibition, engineers will have a good opportunity of comparing the advantages and disadvantages of different types of equipment for specific purposes. Furthermore, they may well find that their attention is drawn to equipment they had not considered but which may be extremely suitable for the purposes they have in mind. This educational aspect of the exhibition is not unimportant.

Trucks

In the automobile industry much movement of material is necessarily carried out by trucking. This industry was one of the first to make wide use of fork lift trucks and many such trucks will be exhibited. *Ransomes, Sims and Jeffries, Ltd., Orwell Works, Ipswich, Suffolk,* will show a range of battery-operated fork lift trucks of capacities varying from 10 cwt at 15 in centres to 2 tons at 18 in centres. Diesel fork lift trucks of 2,000 lb and 5,000 lb capacity will be exhibited by *Coventry Climax Engines Ltd., Widdrington Road, Coventry.* This company will also show the FTX lift truck of 10,000 lb capacity.

An interesting development will be shown for the first time by *Conveyancer Fork Trucks Ltd., Liverpool Road, Warrington, Lancashire.* It is a fork truck with a fully-automatic turbo drive, has neither clutch nor gearbox and has only three driving controls. The capacity is 6,000 lb at 20 in load-centre with a lift height of 12 or 14 ft. Electrically-operated, pedestrian controlled fork lift trucks will be exhibited by *Lansing Bagnall Ltd., Kingsclere Road, Basingstoke, Hants.* The small size and low unladen weight of these trucks make them suitable for use in multi-storey buildings.

For many applications of material handling within a factory, some type of truck other than fork lift is desirable. For some applications, a power-operated tractor for moving heavy loads will be employed, while for others the best solution is to use a manually-operated truck. Where the distances to be moved are short and the loads not too heavy, a manually-operated truck will generally show the greatest economy. For this class of work, *The Yale and Towne Manufacturing Co., Willenhall, Staffs.,* have developed a hand pallet truck that is particularly suitable for use in stores in conjunction with fork lift trucks. It is also convenient for loading road vehicles. Where the movement distance is greater, the products of *Harborough Construction Co. Ltd., Harbilt Works, Market Harborough, Leicestershire,* may prove suitable. They include a pedestrian-controlled electric truck and a rise-and-fall platform truck. The former has a one-ton payload and is operated by a 24 V traction-type battery. It is fitted with a vertically-mounted 2 h.p. motor and has a drive unit comprising a totally-enclosed worm reduction and final chain drive to the front wheels.

Conveyors

It is extremely difficult, in fact almost impossible, to give an adequate display of factory conveyors in the limited space that is available for any one manufacture. Nevertheless, quite a good conception of what conveying equipment is available will be possible. For example, *Fisher and Ludlow Ltd. (Materials Handling Division), Bordesley Works, Birmingham,* 12, will show the whole of the Flow range of handling equipment. It will all be at work. The well-known pressed steel unit-construction Flowline belt conveyor will be demonstrated. In addition to the normal horizontal conveyors, special purpose equipment will also be exhibited. This will include a return belt device, inclined sections, inverse bends and converter units. There will also be a special purpose Flowline conveyor with protected drive and tension units and timber troughs. This is designed for use under conditions for which steel construction would be unsuitable. Other Fisher and Ludlow exhibits include a Flowlink universal overhead chain conveyor with standardized universal fittings; a Flowlink conveyor as a combination of overhead and carousel types; and a Flowlink carousel conveyor incorporating special automatic loading and discharging features.

Many important developments in

mechanical handling have been initiated by *Geo. W. King, Ltd., Hartford Works, Hitchin, Herts.* For example, they recently installed in the new assembly factory of Austin Motors Ltd., a Dual-Duty conveyor system which uses the Hollerith punched card system for selection. A model will be exhibited to demonstrate the principles employed. New light-type Power-Pulled and Dual-Duty overhead conveyors will also be shown working. A working floor-level conveyor with interlacing slats will also be shown.

Engineers concerned with the movement of light loads will find interesting equipment on the stand of *Teleflex Products Ltd., Conveyor Division, Uphall Road, Ilford, Essex.* Three types of conveyors will be shown. One is a dual directional chain unit in the form of a slat conveyor; the second an overhead cable conveyor; and the third a Teleflex torsion conveyor. These equipments are capable of carrying loads varying between 15 and 100 lb/ft, according to type. Teleflex remote controls in their application to industrial purposes will also be shown.

The examples given here of what will be exhibited are far from being comprehensive for trucks and conveyors. They are merely intended to give some slight idea of the wide range of equipment that will be shown in these two ranges. Other types of equipment of interest to engineers in the automobile industry may perhaps not be so much in evidence, but there is little doubt that no matter what the materials handling problem, there will be some suitable equipment on show at this exhibition.

Finally, we would draw attention to the Convention which is concurrent with the Exhibition. So far as the automobile industry is concerned we would draw attention to the following papers:—

Thursday, 5th June, 2.30 p.m. Foundry Mechanization. J. Bain, A.M.I.Mech.E.

Saturday, 7th June, 11 a.m. Materials Handling and the Production Engineer. T. W. Elkington, M.I.Prod.E.

Monday, 9th June, 2.30 p.m. Mechanical Aids in Industry (Mechanical Handling Equipment for Smaller Works). K. B. Warwick, A.M.I.Mech.E.

Thursday, 12th June, 2.30 p.m. Motion Study Applied to Mechanical Handling. L. W. Bailey, F.R.Econ.S., A.M.I.Prod.E.

Friday, 13th June. Mechanical Aids Team speaking on case histories arising from their report.

COMBUSTION IN PETROL ENGINES

THE combustion chamber deposits which are formed after continued operation of petrol engines normally reach an equilibrium thickness owing to flaking off. Carbonaceous in nature with unleaded fuels, they also contain with leaded fuels a substantial proportion of inorganic lead compounds. They are not generally thick enough to cause operational difficulty but substantially increase the octane requirement of the engine. In *Industr. Engng. Chem.*, December 1951, E. C. Hughes and others describe an investigation made to determine whether such increase arises from an undesirable catalytic effect on combustion rather than, as hitherto believed, from an insulating effect of the deposits.

Laboratory tests were made to discover compounds that would inhibit the catalytic action of surfaces of the type in question. The activity was investigated of certain solids on the oxidation of *n*-heptane. Surfaces including porcelain rings, iron oxide, lead oxide, lead borate and lead bromide promoted oxidation between 371 and 427 deg C. Surfaces of lead sulphate, lead chloride and the oxides of boron and silicon had little effect or inhibited the oxidation of *n*-heptane at all temperatures. Engine tests were then made to study the influence of selected compounds on the catalytic effect of deposits with: (1) a modified F-4, single cylinder, aviation fuel knock-rating engine, and (2) a standard 1942 Chevrolet private car engine.

The control test showed, in the octane number requirement for knock-free operation, a rapid rise which levelled off after 80 hr at 12 units above that of the clean engine. With the boron additive, the levelling-off occurred at only 4 units above the octane requirement of the clean engine. Discontinuing the additive after 100 hr gave a gradual increase of 9 more units over a further period of 60 hr. Other tests showed that five out of six boron compounds had a similar effect. A concentration of 3 ml per gallon was found necessary; in the case of an already existing active deposit the effect of the additive was very slow. Tests with silicon compounds and lead sulphate are also reported.

The general results are taken to support a catalytic rather than a thermal conductivity theory as an explanation of the increase in octane requirement caused by deposits. (*M.I.R.A. Abstract No.* 5732.)

GAS TURBINE TRUCK TESTS

IN an *S.A.E. Preprint*, October 29-31, 1951, H. C. Hill reports that in over a year of road tests a standard Kenworth truck fitted with a Boeing 175 h.p., two-shaft, gas turbine unit has covered 15,000 miles with a gross load of 68,000 lb. The absence of throb and cylinder explosions, good "staying" qualities, good response to the accelerator pedal and reduced number of gears, all combined to give very satisfactory driving conditions. Road performance with seven gear steps was approximately equivalent to that of a 200 h.p. Diesel engine with twelve gear steps. The gain is attributed to reduced power expenditure on cooling fan drive, water pump drive, and air cleaner and exhaust losses, as well as to better utilization of peak performance.

With an air inlet silencer and an acoustic muffler on exhaust stacks, the noise level is lower than in standard Diesel trucks, and is capable of further reduction. Consumption and deterioration of the lubricating oil are negligible and the engine is insensitive to the type of oil used. Fuel consumption, at 1 m.p.g. compares unfavourably with the 3 to 5 m.p.g. of modern Diesel trucks. Improvement in consumption is believed possible by: (1) reducing drive-line and transmission losses, (2) reducing engine installation losses, (3) taking fuller advantage of the turbine weight saving, (4) raising the thermal efficiency. In addition, cheaper fuel may be employed.

Major difficulties encountered and solved by improved design included the problem of rubbing contact between the turbine blade tips and the shroud, and blade damage due to foreign objects carried by the gas stream. Numerous failures of the blades were found to be caused by blade and disc vibration, and were eliminated by adjustment in design and fabrication to raise their natural frequencies of vibration. One type of blade cracking was the result of excessive thermal stress accompanying rapid acceleration, and necessitated increasing the idle to full throttle accelerating time from three to five seconds.

Extensive research has produced a burner liner which will operate for 200 hr without any attention and considerably longer with minor maintenance. Difficulty has been experienced in balancing the nozzle area whenever wheel or nozzle replacements were required. Specially developed accessory equipment is described and illustrated. (*M.I.R.A. Abstract No.* 5637.)

JIGS AND FIXTURES

Principles of, and Recent Development in, the "Wharton" Universal System

A SYSTEM of units from which various jigs and fixtures may be simply and rapidly assembled, provided it is adaptable and permits the ready, accurate and rigid locating of components to be machined, has many advantages. The type of fixture best suited to a component can be found, by trial, far more quickly than by the use of skilled labour in preparing drawings. Costly tool-room methods of manufacture, and consequent delay, are avoided. The jigs and fixtures can be adjusted to compensate for wear on dies and patterns, or for modifications in design; and can be prepared to suit any machine in accordance with machine-loading requirements. Production in quantities too small to justify the construction of normal jigs and fixtures can be expedited with economy.

Prototypes can be produced under actual shop conditions, and the assemblies can be used for production until permanent fixtures, if required, have been prepared. Sometimes, the very facility with which jigs and fixtures can be assembled, by unit tool methods, allows greater latitude in the stocks of parts required to be carried. Where a variety of requirements are met, capital charges are low because the constructional units or elements are interchangeable.

The "Wharton" universal jig and fixture system of Wharton and Wilcocks, Ltd., 20, Bull Plain, Hertford, possesses the above advantages. Locational accuracy is ensured by the use of tenons and tenon slots, and by accurately made elements: the tolerance on all major dimensions is 0·0003 in and on squareness is 0·0006 in/ft. Rigidity is served by the use of flat-headed bolts sliding in tee slots and passing through the elements themselves; by the accuracy with which the elements fit together; and by their design and material. Nickel-chromium steel is used for tee bolts and clamps; Meehanite cast iron for the baseplate machined from the solid, seasoned and ground; and steel for other elements, the wearing parts being case-hardened and ground. Across the junction of vertical tee slots in two elements, interlocking tenons may be inserted and secured by Allen screws to give rigidity and ensure alignment of the elements. For simplicity, the sides of the tee slot, where it is provided for bolting purposes, serve as tenon slot for locating purposes. The external size of the elements is so co-ordinated with the standard spacing of tenon slots and tee slots that elements in adjacent slots abut perfectly. In the rectangular baseplates, slots are spaced at unit intervals in one direction and at a whole-number multiple of this interval in the other.

Stop elements are the principal means of building the structure of a jig or fixture. They take the thrusts and stresses. Increased rigidity can be provided by one or more stop thrust elements abutting against any side of a stop element or against each other. They themselves are bolted down either in the same tee slot as the stop element or in an adjacent slot. One thrust element per stop element suffices for the machining of high-tensile steel by negative-rake tungsten-carbide-tipped tools.

Variation in the position of the elements on the baseplate is obtained, in one direction, by movements of the elements along the baseplate slots, and, in the direction of right angles to this, by choice of slot or by use of frame elements (having on the underside a tenon to fit the baseplate, and, on the upper face, a tee slot at right angles to the tenon). The stop elements themselves have horizontal and vertical tee slots. Vertical location is determined by the height of the stop elements or columns of stop elements selected, by the position of the tee slots used for locating purposes thereon, and, within finer limits, by rectangular height elements graduated in tenths of an inch, and shims permitting increments of 0·005 in, or by circular height elements giving vertical increments of 0·001 in. Square height elements are available as extras, but are not included in the basic outfits.

Other forms of frame elements include frames for extending the face area of the baseplate, and frames which can be bolted to, and located in, the vertical tee slots of stop elements, providing a variable height location in that the frame itself has horizontal tee slots in its top and bottom faces. The underside may be supported by height elements.

Angular settings

An important development in the system is the inclusion, in the basic outfit, of angular setting units, the circumference of whose circular bases are graduated in degrees. Their undersides are slotted to take either loose tenons fitting tenon slots in the sides of stop elements or tenon adaptors fitting slots in the baseplate. The upper portion of the angular setting unit rotates in the bore of the lower portion, and itself carries a tenon slot for

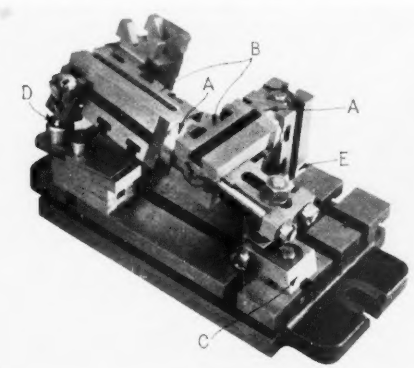


Fig. 1. Compound angle fixture built up from Wharton units

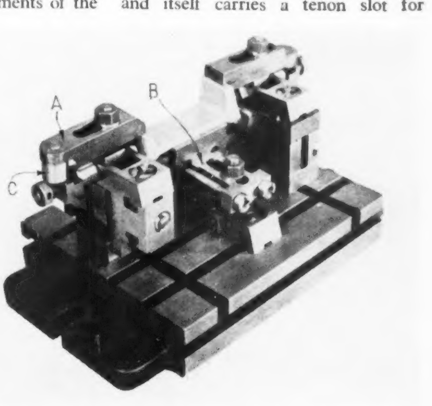


Fig. 2. Angular milling fixture showing clamping methods

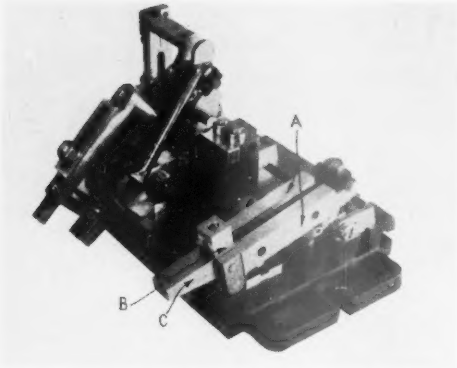


Fig. 3. Showing the application of Wharton sine bar elements

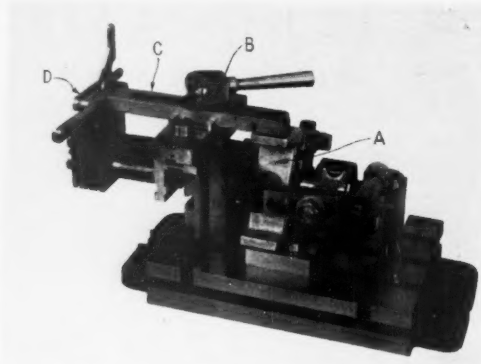


Fig. 4. A fixture for drilling holes at close centres

mounting purposes. These units can be used singly, or in pairs, braced by angle strap or clamp plates.

Fig. 1 shows how two angular setting units *A* can be used to give a simple and a compound angular setting to two stop elements *B*. Fig. 1 shows also a swivel studholder (hook-clamp and swivel) *C* which, attached to the baseplate, takes a stud that can be fitted at the angle required for attachment to the angular clamp plate supporting the stop element positioned at the simple angle. The stop element at a compound angle is anchored to the base through a clamp, universal joint *D*, and jackwedge (for parallel height adjustment) resting on height elements.

Because of the compound angle, the centre line through the vee element at the top left in Fig. 1 is not quite parallel with the tee slot across the right-hand end of the baseplate. Correction for this compound-angle misalignment may be made at the angular setting unit shown at *E* in Fig. 1.

Location by external diameter

For locating components from their external diameters the vee principle is used. The vee elements provided may slide in horizontal or vertical slots of other stop elements. These vee elements cover ranges of diameters up to 2 in. There are also vee blocks designed for use with rectangular height elements and to take up to 3 in diameter. Pairs of half-vee elements can be used for locating any diameter, the half-vees being set apart at the appropriate distance for the diameter to be accommodated.

Bore location is effected by means of a datum at which a ring or pin, turned by user to fit the bore, can be mounted. The datum is provided by a short, ground cylinder, known as a location unit spigot. It may be either 1 in or 1½ in diameter, and has a tenon on one end so that it may be fitted to the tee slot of a stop element or baseplate. A bolt is passed through the centre of the spigot to hold the ring. For small diameter bores, a special location unit adaptor can be screwed in to the base of the location unit. The adaptor itself has a No. 1 Morse taper ½ in diameter in which the locating pieces made by the user are screwed. The combined depth of the location unit plus the adaptor is 1 in.

Clamping

Close consideration has been given to the means for clamping components for machining. For milling processes, jack screws are supplied for use in conjunction with jack bodies or clamps. There are also pillar-jack screws, slotted clamps, swing-over clamps and guides. There are also jack-wedges and the equipment shown in Fig. 1 for supporting stop elements at simple and compound angles. Fig. 2 shows an angular milling fixture with a clamp *A*, a clamp plate *B*, and a clamp heel *C*. Clamp heels are made in both brass and steel, with or without extension, and may be screwed to the clamps . A ball-ended clamp heel located in a socket screwed into a plain clamp is required for clamping components on to angular faces.

Bush holders for drilling and boring operations may be of either split-grip or solid-end type. Split-grip holders have bores to take British Standards press-fit bushes. A set-screw is used to tighten the grip. Solid-end bush holders are designed to take slip bushes. The face is tapped to receive an anchor pin. Each type of bush holder has a slotted shank for bolting purposes and slides in a sliding base that is attached by tenons to the slot of an angle strap or other element.

Mountings can be doubled. An anchor plate is attached at the end of the bush holder remote from the bush, and endwise locking, or fine positioning, of the bush holder is by locking nuts on studs passing through the anchor plate. To allow the bush holder to be swung clear for loading and unloading components, a hinged sliding base is available. If holes have to be drilled at close centres, two or more hinged mountings can be used, so that only one is in the working position at any one time.

Additional equipment

There are several items of equipment that are not supplied with the basic outfits. They include sine-bar equipment, see Fig. 3, for accurate angular settings. In Fig. 3 a hinged sine bar is shown at *A*, a clamp adaptor at *B* and a stop at *C*. There are also knife holders for positive endwise location of bush holders on the knife-edge-in-slot principle. This arrangement is shown in Fig. 4, which is a drilling fixture designed for drilling holes at close centres. The component is shown at *A*, a cam clamp at *B*, the bush holder at *C* and the knife holder at *D*. Other extra equipment includes a circular table for mounting on a standard base, or on the back plate of a lathe. It is graduated round the circumference to facilitate accurate indexing.

Wharton units are supplied in three series, of which the medium size with $\frac{7}{16}$ in and $\frac{9}{16}$ in slots is the most popular. Large and small series have $\frac{11}{16}$ in and $\frac{9}{16}$ in, and $\frac{7}{16}$ in and $\frac{5}{16}$ in slots respectively, so that the medium series is partially interchangeable with both the others. The complete range of elements is great enough to provide for the assembly of almost any type of jig or fixture.

Radioactive Isotopes

A HANDBOOK published by the U.S. National Bureau of Standards contains the best advice currently obtainable on the minimal precautions to be taken in the disposal of the isotopes phosphorus-32 and iodine-131. Presuming that disposal will be by sewer it considers permissible concentrations to ensure the safety of sanitation workers in particular and the community in general. Priced at 10 cents, copies of this publication (Physics H49) are obtainable from the Superintendent of Documents, U.S. Government, Printing Office, Washington 25, D.C. (1994)

THE 4·1 LITRE COMMER ENGINE

A Vertical O.H.V. Unit for the Superpoise Range

SPECIFICATION

Number of cylinders	6
Bore and stroke ..	$3\frac{1}{2}$ in × $4\frac{3}{8}$ in (88·9 mm × 111·1 mm)
Swept volume ..	252·6 in^3 (4,138 cm^3)
Compression ratio	6·48 : 1
Maximum b.h.p. ..	85 at 3,200 r.p.m.
Maximum b.m.e.p.	120 lb/in^2 at 1,200 r.p.m.
Maximum torque	2,400 lb-in at 1,200 r.p.m.
Crankshaft	7 bearings. Integral balance weights
Valves	Overhead, push rod operation
Carburettor ..	Solex 35AIP

SHORTLY before the War, the Superpoise range of vehicles, powered by a 4,086 cm^3 side valve engine, was introduced by Commer Cars Ltd. of Luton, and remained in production until last year. In 1951 improvements to this range were announced. These included some alterations to the chassis, but the most outstanding modification was the incorporation of a new vertical, six cylinder, overhead valve engine of 4,139 cm^3 capacity. As before, a Perkins diesel unit can be installed as an option.

The new Superpoise engine is a development from the 4,750 cm^3 six cylinder O.H.V. underfloor unit, described in the February 1949 issue of *Automobile Engineer*. Many of the components are common to both engines, and a number of the crankcase machining operations are identical. An increase, as compared with the side valve unit, of 7 per cent in b.h.p. and 14 per cent in torque has been effected. Another improvement over the earlier Superpoise engine is the employment, as in the underfloor unit, of a seven bearing crankshaft.

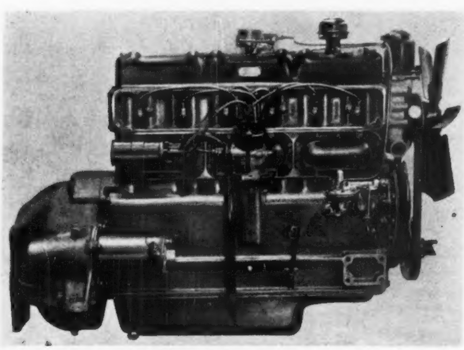
So far as general layout is concerned, the engine is fairly conventional. At the rear, in a cast iron pot-type housing, is an 11 in diameter Borg and Beck 11A6 clutch. The housing is bolted to the crankcase by two set bolts at the top, and six nuts and bolts at the sides. Access to those at the sides is from the inside, so that the crankshaft, flywheel and clutch must be assembled as a unit into the engine after the housing is fitted to the crankcase. A two-piece pressed-steel cover is bolted to the lower face, and integrally cast brackets form the rear engine mounting. The Lucas M45G starter motor is bolted on the right-hand side. A dowel is screwed into the front face of the housing on the left-hand side. It can be reversed and engaged in holes in the flywheel to set the engine for timing.

Also on the right-hand side, but approximately mid-way along the crankcase wall, is the ignition contact breaker and distributor, with its centrifugal and vacuum-controlled automatic advance and retard mechanism. The unit is supplied through a Lucas B12L-O coil mounted on the scuttle. An oil level dipstick is carried in a boss low down on the crankcase wall. Three pressed-steel tappet covers are bolted to the cylinder block, and on the rear one is welded the engine breather pipe. The engine ventilation system is completed by another pipe, from the top of the rocker cover to the air intake. Above the tappet covers are the spark plug bosses recessed into the cylinder head.

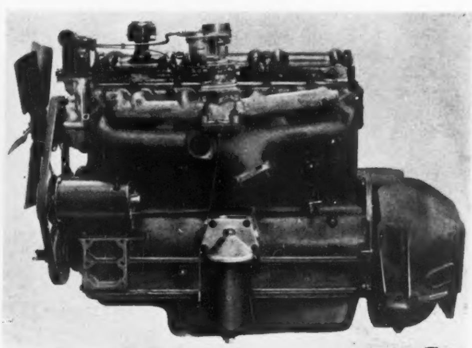
On the left-hand side are the manifolds and carburettor. Below these, bolted to the crankcase, is a full-flow A.C. oil filter which is a fairly recent addition and is not in the illustration of the general arrangement. To the rear of the filter is a water jacket drain cock, and in front of it bosses are provided for the bolts securing the pivot bracket for the Lucas C45 PV-4 dynamo, and for the engine mounting brackets. The dynamo is driven, together with the fan and water pump, by the usual triangulated V-belt drive. The water pump is bolted to the front end of the cylinder head.

Cylinder block and crankcase

The integral cylinder block and crankcase is of cast iron. There are five crankcase webs which, together with the front and rear end walls, support the crankshaft journal bearings. To the right of these, bosses in two of the webs and in the end walls carry the four camshaft bearings. With such closely spaced webs it is unnecessary to incorporate any further ribbing to stiffen the crankcase walls that extend to a level $3\frac{1}{2}$ in below the axis of the journal bearings. In this crankcase, long enough to accommodate a seven-bearing crankshaft, there is ample water jacket space around each of the cylinders, the walls of which are $\frac{1}{4}$ in thick. The minimum space is $\frac{1}{2}$ in, and that between the centre pair $1\frac{25}{32}$ in. The cylinder



When the engine is installed in the vehicle, the coil is mounted on the scuttle



A dowel, screwed into the flywheel casing, may be reversed to locate the flywheel for timing the engine

bores are not finished with chromium as were those in the earlier underfloor engines.

At the rear of the unit is a flange for the clutch housing. On the front end of the crankcase, the oil seal is formed by a machined face to which is bolted a front engine plate, ¼ in thick, and the pressed-steel timing cover, both joints being made with Oakencork washers.

Lateral location of the cast iron bearing caps is effected by recessing them $\frac{5}{16}$ in into the webs and end walls of the crankcase. To ensure assembly the correct way round, their recessed flanges are a different thickness on each side. All on the right are flush with the lower face of the web while the thicker ones on the left project below it. In addition, their ½ in diameter En 16T studs, used in conjunction with self-locking nuts, are asymmetrically positioned about the axis of the bearings.

At the front bearing cap, the oil seal is formed by a cast iron bridge-piece secured to the cap by countersunk set screws. On each end-face of the bridge-piece there are diagonal grooves in which seat cork strips to bear against the sides of the housing in the crankcase. The ends of the grooves are respectively at the front face and the sump joint-washers.

At the rear bearing cap a similar type of seal is used. The abutting faces, however, are assembled with shellac, and hard felt sealing strips are employed. This type of strip has been found necessary because the cork ones are liable to shrink in store and, of course, there must be no possibility of even a slight leakage of oil into the clutch housing.

Vandervell D2 Bimetal, steel-backed micro-babbit, journal bearing shells are fitted. They are located axially and held against rotation by means of tongues pressed out at the abutting face of each half. These engage in slots machined in the cap and housing. The length of the front, centre and rear bearings is $1\frac{3}{8}$ in, while that of the intermediate ones is $1\frac{5}{16}$ in. All main journals are $2\frac{3}{4}$ in diameter.

Crankshaft, connecting rods and pistons

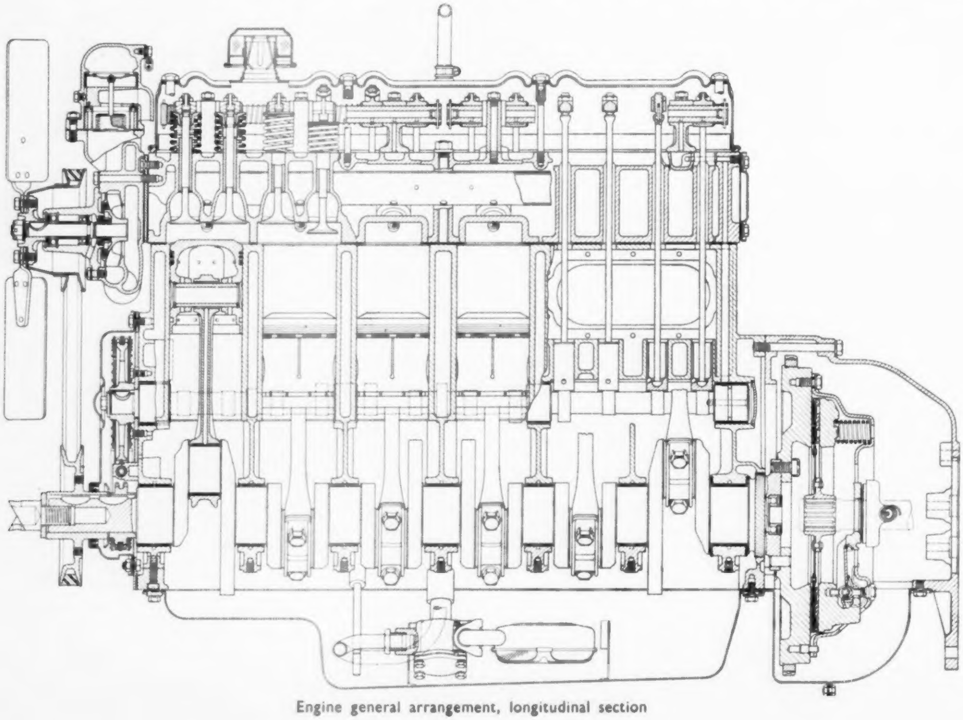
A drop forged, En 10 crankshaft is employed. Its length, between the front of number 1 bearing and the back of number 7, is $29\frac{13}{32}$ in, and the crank webs are $\frac{11}{16}$ in thick. Four balance weights are formed integrally with the webs; one on each side of the centre bearing to relieve the load on it, and one on each of the webs adjacent to the outer face of numbers 2 and 6 bearings, to balance the centre pair of weights.

End location of the crankshaft is effected at the rear journal bearing, where Vandervell D2 Bimetal, steel and babbit, divided thrust washers are positioned each side of the bearing and cap. A locating tab is formed on each lower half of the washers. On assembly, it is pressed into a slot in the appropriate face of the bearing cap.

The thrust is taken on the front washer by the crank web, and on the rear one, by a shoulder and oil thrower machined on the shaft. Immediately to the rear of the oil thrower is a machined flange $\frac{15}{32}$ in wide, with oil return grooves cut at an angle of 45 deg on its periphery. They work in a boss in the rear crankcase wall.

Six $\frac{7}{16}$ in diameter bolts, locked by tab washers, carry the cast iron flywheel on a flange formed on the tail end of the crankshaft. A dowel ensures positive location. Shrunk on the rim of the flywheel is the starter ring-gear of hot-rolled En 8D strip. The clutch mainshaft is supported in a sealed ball bearing housed in the bored end of the crankshaft.

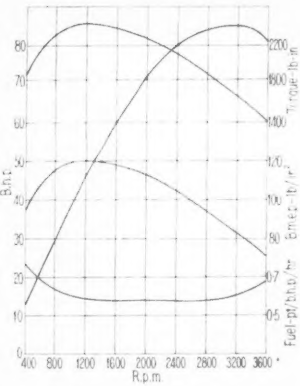
Drop forged, H-section connecting rods, of En 16S, have a centre-to-



Engine general arrangement, longitudinal section

centre length of $8\frac{1}{4}$ in. The big ends are split at an angle of 45 deg so that the rods may be withdrawn through the cylinder bores. Vandervell D2 Bimetal, steel-backed, micro-babbit, shells are $2\frac{1}{4}$ in diameter and $1\frac{11}{32}$ in long. Two $\frac{7}{16}$ in diameter set bolts, of En 19C, secure the bearing caps, positive location being effected by dowel tubes, $\frac{13}{32}$ in long, around the bolts. The bolts are locked by tab washers. This is a departure from previous practice; in the first underfloor engines they were wire-locked, and in the side-valve engines, split pins were used. Pressed in the small ends are Clevite 10 lead-bronze bushes, $1\frac{1}{4}$ in long. They are positively lubricated through an axial drilling from the big end. The inside and outside diameters of the hollow gudgeon pin are respectively $\frac{25}{32}$ in and $1\frac{1}{8}$ in. Seager circlips, in the piston bosses, axially locate the gudgeon pin. Splash lubrication is assisted by a vertical drilling in each boss.

Lo-Ex die-castings for the T-slotted pistons, are supplied by the Northern Aluminium Co. Ltd, and the machining is done by the engine manufacturers. Two compression rings and one slotted scraper ring, all made by Hepworth and Grandage Ltd.

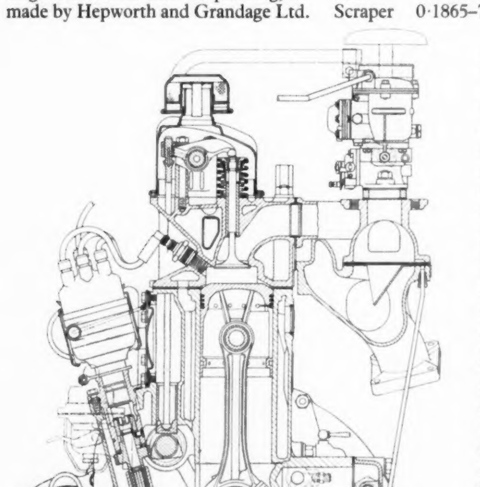


Engine performance curves

are fitted above the gudgeon pin. The top ring is chromium plated. In the past both compression rings have been of plain rectangular section but it is intended that in future they will have a No 2 taper on their faces. The ring dimensions are :

	Face width	Radial thickness
Top	0·0928–38 in	0·138–40 in
2nd	0·0928–38 in	0·138–40 in
Scraper	0·1865–75 in	0·134–40 in

There is a ring groove in the skirt so that an extra scraper ring may be fitted later in the life of the unit to reduce oil consumption and extend the period of operation of the engine before a rebore is necessary. The piston crown is shaped like a shallow truncated cone. This somewhat unusual shape has been adopted in order that the cylinder head may be common to this engine and the larger bore underfloor engine. The piston crown projects into the combustion chamber in the cylinder head to give a compression ratio of 6·48 : 1.


Engine general arrangement, transverse section

Camshaft and valve gear

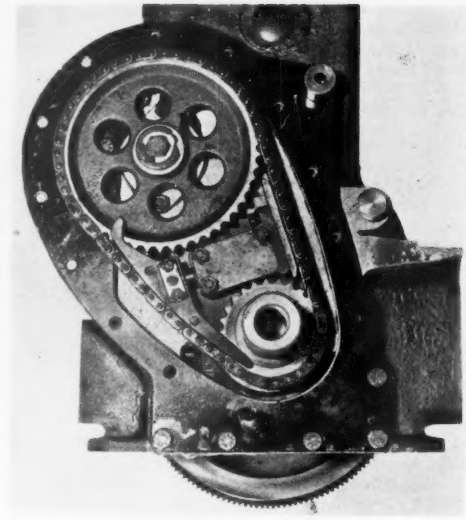
Both the drop forged En 5C, driving sprocket for the timing chain and the cast iron, fan-belt drive pulley are carried on the $1\frac{5}{8}$ in diameter front extension of the crankshaft. Between them is a pressed steel, oil thrower ring, and immediately in front of this, bearing on a rearward extended boss of the fan-belt pulley, is a Gaco oil seal in a housing pressed in the 16 S.W.G. steel timing cover. Two tapped holes, for an extractor, are provided in the pulley wheel, which is a push fit on the shaft. The whole pulley and sprocket assembly is pulled up against the front face of number 1 journal by a $\frac{1}{4}$ in thick washer and a special bolt incorporating the dogs for the starting handle. Each of the wheels is driven by a separate Woodruff key. Machining the keyways is, of course, a quicker operation than in the earlier engines in which one long key drove both wheels.

A two-row roller chain of $\frac{3}{8}$ in pitch transmits the drive to the half-speed wheel. Tension is maintained in the chain by a blade-and-slipper assembly carried on a spring-loaded plunger. The cast iron body of this assembly is secured by three $\frac{1}{4}$ in diameter set bolts to the front wall of the crankcase, between the camshaft and the crankshaft. Bearing, as it does, against the inner face of the chain, it is assisted by centrifugal force in maintaining the tension when the engine is running. The $\frac{1}{2}$ in diameter bore in the body is closed at one end by a cold drawn En 1B steel plunger, and at the other end by an En 2 slipper. The inner end of the plunger is drilled axially to carry a compression spring which bears against the slipper. Oil, splashed off the half-speed wheel, passes through a drilling at the top of the body to lubricate the plunger sliding in its bore. The slipper is held in position by two countersunk set screws, one in a tapped hole in the body and the other being secured by a nut. With the chain taut, there is a clearance of $\frac{3}{64}$ in between it and the slipper.

Pivoted to the outer end of the plunger is the blade assembly. This is built up of nitrided En 41 laminations. The outer pair are $\frac{3}{16}$ in thick, and they bear on the two outer rows of chain links. Four more, smaller laminations space the outer blades $\frac{21}{32}$ in apart. The whole assembly is copper brazed together being held for this operation by three $\frac{1}{8}$ in diameter rivets. Projections on the outer blades are drilled to carry the pivot pin. This has an annular groove at one end so that it may be located by a U-plate, registering in the groove and secured to the front blade by two tab washered $\frac{3}{16}$ in diameter set bolts.

The front end of the camshaft is drilled and tapped for a $\frac{3}{8}$ in diameter set bolt securing the cast iron half-speed wheel driven by a Woodruff key, there being no provision for fine adjustment of the valve timing. The bolt is tightened against a washer, $\frac{1}{8}$ in thick, which pulls the sprocket up against a shoulder on the shaft. Immediately behind the shoulder the diameter of the shaft is again stepped up to form the front bearing. Axial location of the camshaft is effected by an En 2B, ground and hardened plate, secured to the crankcase by two $\frac{1}{4}$ in diameter set bolts. It registers around the shoulder between the front bearing and the boss of the half-speed wheel. To facilitate assembly, the four bearings of the drop forged En 32B camshaft are stepped down in diameter from $1\frac{7}{8}$ in at the front to $1\frac{3}{4}$ in at the rear. Pressed into the bearing bosses in the crankcase are the Vandervell steel and babbit, D2 Bimetal, camshaft bearing bushes. Their lengths are: front $1\frac{1}{2}$ in, rear $1\frac{19}{32}$ in, intermediate $1\frac{3}{32}$ in. A pressed-in, 12 S.W.G., steel disc seals the rear bearing boss. To prevent the disc from being forced out, the oil pressure is relieved by an axial drilling in the end of the camshaft, which communicates with a radial one just in front of the bearing. Half way between the front two bearings is the fuel pump actuation cam. Similarly positioned between the intermediate pair of bearings is a spiral gear to drive the spindle for the oil pump, and the contact breaker and distributor. The cam contours are designed for smooth valve operation, and incorporate ramps for taking up the tappet clearance quietly.

The $\frac{9}{32}$ in diameter, En 34, push rods have an effective length of $12\frac{7}{16}$ in. Their ends are up-set and case-hardened. At the bottom they are carried in spherical seatings in the Brico, chilled cast iron, piston-type tappets, and at the top they are cupped to seat the ball ends of the En 351 tappet adjusting screws. These are carried in the rocker ends, and are fitted with lock-nuts. Brico cast iron rockers with chilled end-pads are employed, the rocker ratio being 1·295 : 1.



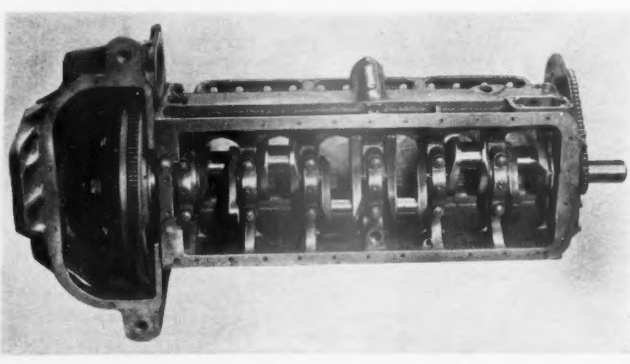
The blades of the tensioning device bear on the side plates of the timing chain

VALVE DATA

	Inlet	Exhaust
Material	En 52 steel	X.B. steel
Head diameter	1·745 in–1·741 in	1·515 in–1·511 in
Throat diameter	1·50 in	1·378 in–1·372 in
Stem diameter	0·3735 in–0·3728 in	0·3735 in–0·3728 in
	inner—outer	inner—outer
Spring rate	56·7 lb in–91·5 lb in	56·7 lb in–91·5 lb in
Spring length free	2·16 in–2·44 in	2·16 in–2·44 in
Spring length installed	1·60 in–1·75 in	1·60 in–1·75 in
Valve lift	0·41 in	0·41 in
Tappet clearance	0·012 in	0·014 in
Valve opens	15 deg B.T.D.C.	57 deg B.B.D.C.
Valve closes	57 deg A.B.D.C.	15 deg A.T.D.C.
Ignition timing (static)	2–4 deg B.T.D.C.	

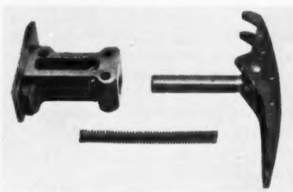
In order to facilitate machining, by reducing the length, the En 43B hollow rocker shaft is in two pieces, each $13\frac{1}{2}$ in long. The inside and outside diameters are respectively $\frac{5}{16}$ in and $\frac{13}{16}$ in, and pressed-steel cups seal their ends. Each shaft is supported on three pedestals, situated one between each pair of valves, and centred $1\frac{3}{4}$ in from a line through the valve axes. These pedestals are made of STA7 AC4 cast aluminium so that there is no possibility of the tappet clearances closing as the engine warms up. To avoid the need for fitting separate caps, they are bored. In the rear one supporting each shaft, slots parallel to the axis and about $\frac{5}{16}$ in deep, are cut, in the horizontal plane, diametrally opposite one another in the bores. Thus, when the nuts are tightened on the two $\frac{5}{16}$ in diameter studs securing each pedestal to the head, they clamp the rocker shaft and so locate it axially, as well as against rotation. The rockers are constrained against the pedestals by compression springs between cold-rolled, mild steel washers, located around the spindle, Secured by a split pin on the front and rear of each shaft is a collar against which the end springs bear.

With each valve, two concentric valve springs are employed, the inner and outer ones being coiled in opposite



The complete crankshaft, flywheel and clutch assembly is fitted into the crankcase and clutch housing

directions to prevent interlocking. At their lower ends, the inner spring bears on a pressed-steel locating washer, and the outer one on the cylinder head. The springs are secured and located at their upper ends by split collets and a stepped washer. The washers for the inlet valves each have a pressed-steel shroud around their central spigot inside the coil springs. Although the



An oil chamber is formed in the rear face of the timing-chain tensioner body

shrouds are not necessary on this engine, they are fitted so as to be interchangeable with those on the underfloor engine where, being in a horizontal position, they are required to prevent excessive quantities of oil from draining down the rockers on to the valve stems. Oil drainage from the split collets of the inlet valves is restricted by a rubber ring in a groove around each stem. Details of the valves, springs and timing are given in the table on the preceding page.

The cast iron valve guides, vertically mounted, are $2\frac{5}{8}$ in long by $\frac{5}{8}$ in outside diameter, and are interchangeable. They are chamfered at the bottom so that a sharp edge is formed around the inner periphery. This, in conjunction with the step, or undercut, around the valve stem at this end, helps to clear away any carbon that might form.

Cylinder head and manifolds

An open-ended, cast iron cylinder head is employed so that the core may be removed easily after casting. Inserted into the head through the front end is a B.S. STA7/CZ4B brass tube, extending the whole length of the block. The circular section front end of this tube is carried in the $\frac{1}{8}$ in thick front sealing plate, to which is mounted the water pump. At its rear end, it bears against the pressed-steel cover sealing that end of the head. The tube is loosely supported along its length by the walls of the water jacket. Holes are drilled in its sides to direct cooling water around the valve guides and seats, and around the bosses for the long reach Champion NA8, 14 mm, spark plugs.

The combustion chambers are the same on this engine as in the underfloor unit, except that the lower face of the head is recessed to take the conical piston crown. This has resulted in the combustion chamber being in a rather high position in the head, with the valves placed vertically above it, and has made the casting $4\frac{1}{4}$ in deep. Accordingly, the designers have been able to incorporate generous cooling passages, and the head is particularly resistant to distortion. It is secured to the cylinder block by fourteen, $\frac{7}{16}$ in diameter, En 16T studs with nuts and spring washers, the joint being made with a copper and asbestos gasket. Four internally threaded bolts on studs screwed in the head pull down the pressed-steel rocker cover on cork joint washer of $\frac{5}{16}$ in square cross section.

Although the exhaust port is turned through an arc of small radius, the axis of the inlet port turns through 90 deg with a radius of about $2\frac{7}{16}$ in. Projecting very little into the ports, the inlet guides are well faired, so as to present as little obstruction to the gas flow as possible. The inlet manifold is spigoted on sleeves inserted in numbers 2 and 5 ports at the face on the cylinder block. This ensures that the passage between the two components is in accurate alignment.

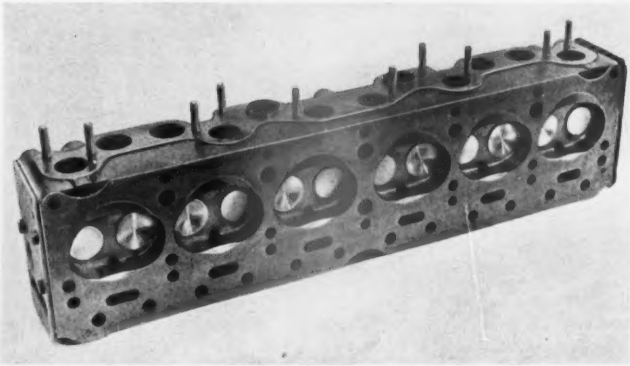
All cylinders are individually ported. On top of the STA7/AC4 cast aluminium inlet manifold, a Solex 35AIP carburettor is bolted. Below, at its junction with the cast iron exhaust manifolds, a 16 S.W.G. stainless steel plate with a welded-on deflector is fitted. The joint is made gas-tight with thin corrugated copper joint washers, one on each side of the plate. From this point, the lowest in the induction system, a drain-pipe extends down to the level of the sump face-joint. At its lower end, a simple non-return ball valve prevents air from leaking into the manifold and upsetting the proportions of the slow-running mixture. The two manifolds are secured at each end by a bolted-on stirrup, but the intermediate ports are individually secured by means of nuts and studs on the block.

Water pump and cooling system

Bolted on to the water pump body, of STA7/AC4 cast aluminium, is a cast iron thermostat housing and water outlet. The front cover of the pump, also made of cast iron, houses two ball bearings, the inner races of which are spaced $1\frac{1}{16}$ in apart by a distance tube. Carried in these bearings is the En 43B pump spindle. On the front end of the spindle is a split-pinned nut and a washer by means of which the cast iron hub for the fan with its two-piece fabricated pressed-steel pulley is secured, the drive being taken through a Woodruff key. The whole assembly, comprising the hub, the two bearings and their distance piece, is pulled up against a collar machined on the spindle. Axial location of the assembly is effected by the outer race of the front bearing, which is housed between a circlip and a shoulder in the pump cover. The outer race of the rear bearing floats axially in its housing.

Mounted on the spindle, behind the collar, is a thrower ring working in a drainage space. To the rear of this, is the water seal supplied by the Morgan Crucible Co. Whereas in the first underfloor engines, the seal rotated with the impeller, in this unit it is stationary, and its moulded-in carbon ring bears against the impeller, instead of against a thrust washer on the rear face of the front

Two rocker shafts are employed, each being a convenient length for machining



The inlet manifold is located by sleeves in numbers 3 and 4 ports on the cylinder block

cover. The cast iron impeller is, as before, pressed on the rear end of the spindle, and it is furnished with tapped holes for an extractor.

Coolant passes from the pump into the front end of the tube directing it around the hot parts of the head. A subsidiary thermo-syphon action cools the cylinder walls, large ducts being provided for the passage of water between the cylinder head and block. The return circuit is through the thermostat valve to a finned tube radiator of 426 in^2 frontal area, served by a six-bladed, fabricated fan. Drain cocks are provided in the cylinder block, and in the base of the radiator.

Oil pump and lubrication system

The gear-type oil pump is now submerged in the pressed-steel sump of 14 pints capacity, instead of being carried externally on the side of the crankcase as it was in the side-valve and the underfloor engines. The idler gear is carried on a short spindle pressed into the cast iron pump body. Meshing with it is the driving gear, keyed on the lower end of an En 43B driving spindle. The bolted-on bottom cover of the pump locates the two cast iron gears in the casing.

An upper extension of the pump body, about 10 in long, is bored at its lower end to form a 1⅛ in long bearing for the driving spindle. In a countersunk hole in the side of this upper extension seats a conical-ended bolt, screwed in from the outside of the crankcase, to locate the pump vertically. Pressed into the top end is a D2 Bimetal bush. It carries the lower boss of the En 32B spiral gear that engages with a driving gear machined on the camshaft. Splined into this boss is the upper end of the pump drive spindle.

The upper boss of the spiral gear, bearing in a cylindrical cast iron adaptor carrying the contact breaker and distributor unit, has a slot in its end, and an annular groove around its periphery. The slot engages the tongued driving sleeve of the contact breaker and distributor, the spindle of which is supported in the boss. A loose peg in the adaptor engages in the groove to prevent the gear from moving downwards out of mesh, should the engine backfire. Around the adaptor is a circlip bearing against the end of the peg to stop it coming out of its hole. The normal direction of the rotation of the driving gear on the camshaft is such that it tends to lift the driven gear to bear against a phosphor bronze thrust washer interposed between it and the adaptor.

In the oil pump, as well as a radial drilling through the idler gear which intermittently contacts a longitudinal flat on its spindle, there is a subsidiary lubrication circuit. This supplies the upper bearing surfaces of the driving spindle which in so many designs, are independently served with oil. A groove in the top face of the driving gear in the pump passes oil to a spiral groove around the periphery of the lower end of the spindle. This carries the oil into the space, around the spindle, in the upper extension of the pump body. It then goes through a radial drilling connecting with a short axial one, in the top end of the spindle, to the interior of the spiral gear boss. From there, it is fed through a radially drilled duct to the thrust bearing, and passes out to help lubricate the gear teeth. Another radially drilled passage in the boss passes oil into the groove for the locating peg, and lubricates the bearing in the adaptor. The top end of the hollow boss is sealed by a pressed-in, cup-shaped plug. Above the boss is a hole, in the side of the adaptor, through which oil, passing upwards from the bearing, may drain to the sump.

For the main lubrication system, oil is drawn to the pump through a floating separator and strainer of 5¼ in diameter. From the pump, a pipe of ⅝ in outside diameter carries the lubricant to a union on an adaptor screwed into the crankcase at the lower end of a ¾ in diameter vertical drilling. This is joined at its upper end by a transverse one directing the oil through a relief valve, set at 55-65 lb/in^2, into a full flow A.C. oil filter. It returns from the filter into a 1¼ in diameter lateral drilling that communicates with a gallery in the crankcase wall. This unusually large diameter drilling is incorporated because, in the underfloor engine, it houses a part of the oil pump body. In the crankcase webs, lateral ducts pass the oil from the gallery into annular grooves around the main bearings, whence it goes through four holes in each shell bearing to lubricate the working surfaces. Drillings in the crankshaft carry oil to the big ends.

From the duct connecting the gallery to the front journal bearing there is a drilling, in the front end of which is screwed a metering jet. This delivers oil into a chamber formed between the base of the timing chain tensioner body and the front wall of the crankcase. Another drilling in the body feeds oil into the meshing point between the chain and driving sprocket.

All the camshaft bearings are supplied with lubricant by ducts from the grooves around the crankshaft journals. Machined around the rear camshaft bearing is a deep annular groove. It is connected intermittently, by a flat on the bearing, with a vertical drilling through the cylinder head to the rear rocker pedestal. From here the oil passes through the shaft, down a duct in the front pedestal, and through drillings in the head to the rear pedestal of the front portion of the shaft into which it is fed. Radial drillings distribute oil to each rocker bearing. Ducts, in the rockers and tappet adjusting screws, positively lubricate the bearing face at the push rod ends, and a leak hole on the other side permits a small amount of oil to run down to the valve ends.

Colloidal Graphite

A BOOKLET newly published by the manufacturers of Oildag gives information regarding the use of colloidal graphite in oil as a running-in compound, as an upper cylinder lubricant, and as an additive to penetrating oil.

In addition to general engineering applications, it also briefly refers to the value of graphite in machining processes. Copies of the booklet may be obtained from Acheson Colloids Ltd., 18, Pall Mall, London, S.W.1. (2009)

CHROMIUM PLATED PISTON RINGS

A Résumé of Experience with Vacrom Rings

ALTHOUGH to-day many manufacturers of internal combustion engines fit chromium plated rings as standard practice, in many quarters there is still insufficient realization of the great improvement that such rings can effect in engine wear. Originally, chromium plated piston rings were fitted to minimize ring wear, particularly under desert conditions. Rather surprisingly, it was found that not only was ring wear reduced, but in addition there was also a considerable reduction in cylinder bore wear. In fact, the effect on bore wear is now the primary reason for fitting chromium plated rings.

Why these rings should greatly reduce the rate of wear in the cylinder bore is still a matter of conjecture. However, there is no doubt that with the correct type of plating and with proper adhesion to the base cast iron ring, chromium plated rings do lead to a great decrease in cylinder bore wear. It must, however, be stressed that special manufacturing techniques must be employed if optimum results are to be obtained. These notes, which are based upon the experience gained by Hepworth and Grandage Ltd., St. John's Works, Bradford, in the development of Vacrom rings. They also include actual service results with such rings in different types of vehicles.

Incidentally, before the introduction of chromium plated rings, chromium plated cylinder liners were tried as a means of reducing the rate of bore wear. These were relatively expensive, and in addition, experience showed that with a hard chromium surface in the bore, lubrication was unsatisfactory. When, however, the hard chromium is applied to the relatively small surface of the piston ring and it is run on a comparatively very large oil-bearing surface of dissimilar material, there are no difficulties in obtaining adequate lubrication.

Manufacturing technique

The first problem that had to be solved before consistently good chromium plated piston rings could be produced, was that of the deposition of the chromium on to the cast iron ring. Chromium can be applied to any type of cast iron, but there is a wide variation in the plating susceptibilities of irons with different compositions, and lengthy investigations were necessary to establish the plating procedures to be employed with different irons. Actually, most rings for chromium plating are made from high duty iron of the "free carbide" type developed for aircraft engines. The use of this type of material is advisable since it gives reduced side wear and freedom from collapse through exposure to high temperatures, such as are experienced in diesel engines.

It is desirable to have the plate as hard as possible, and it is essential that the adhesion between the plate and the base rings should be such that it is impossible to separate one from the other by any mechanical means such as hammering or breaking. To ensure this degree of adhesion the cast iron ring is etched before it is plated. This produces a surface into which the chromium can key itself with great tenacity. The method of etching and the period for which the work is in the etching medium are very important in the effect they have on adhesion. They are, therefore, strictly controlled. Here it should be stressed that there is a great difference between the deposition of plating on a piston ring and deposition on such articles as bumper bars and door handles. Where chromium is used mainly for decoration, even on exterior work the thickness of deposit is in the order of 0·00025/0·0004in whereas for piston rings in petrol engines it will be in the order of 0·0025/0·003in and for diesel engines 0·004/0·006in. The necessary great adhesion can be obtained for thicknesses of these orders only through very close control of the plating process.

When chromium is deposited, it is impossible to avoid a heavy build-up at sharp corners. For this reason rings cannot be plated individually. They are assembled into a long stack of rings and all the joints are filled. Even so, if sharp corners are used, ragged edges will be left when the rings are separated. To overcome this, the outer corners are chamfered before plating. This causes the plate to lap over the corners and taper away down the side. It also helps to increase adhesion.

Bedding-in

Unfortunately, any step taken to prolong cylinder and ring life, also extends the period necessary for running-in. Special precautions must, therefore, be taken with chromium plated rings. One method is to lap the full face of the ring to get perfect contact. This has two drawbacks. First, the ring must still bed into the cylinder, which is seldom truly cylindrical, and during the period it remains unseated, blow-by and oil consumption may be heavy. Second, in the lapping operation it is difficult to avoid the production of a barrelled face on the ring, and such a condition further worsens oil control.

Better results have been obtained by turning the ring with a taper of from ½ to 1 deg on the periphery before plating. The ring is then lapped to produce a very narrow land round the periphery, which gives almost line contact with the cylinder bore. Such rings seat very quickly and also give better oil control during the initial stages. Furthermore, whereas parallel rings have been known to stick after a few hours' operation owing to blow-by, taper-face rings are completely free from this trouble.

Standard practice at present is to use a plate thickness of 0·003in for petrol engines and 0·005in for diesel engines. In general, plating of these thicknesses will be worn through at about one-half of the cylinder life. To make the plate thick enough to last the full life of the cylinder would entail a disproportionate rise in cost. In any event, it is good practice to draw the pistons at half mileage for cleaning and renewal of scraper rings. Unless this is done the oil consumption tends to become excessive. Originally, some doubt was felt about how a new plated ring would bed in a worn bore, but experience has shown that a taper faced ring can be installed into a worn bore without trouble, and will quickly settle down.

Service results

In petrol engines the rate of cylinder wear when chromium plated rings are fitted is usually about one-quarter of the rate when normal rings are fitted. This is irrespective of operating conditions. Thus, when every precaution is taken to ensure cleanliness and correct operating temperatures, and there are few cold starts, rates of wear in the order of 0·001in per 40,000 miles are possible with chromium plated rings. Conversely, under the worst conditions where abrasive can enter the engine and the jacket temperatures are low, the proportionate wear remains the same.

For reasons not clearly understood the improvement effected by fitting chromium plated rings in diesel engines is not so marked. Under similar conditions, a normal cast iron ring will frequently last longer in a diesel engine than in a petrol engine, but a chromium plated ring will last only half as long in a diesel engine as it will in a petrol engine. In other words, the ring: bore life ratio as between chromium plated and normal rings is 2:1 for diesel engines and 4:1 for petrol engines. Some actual results are appended.

Case A. A large fleet of small 8 h.p. vans, operating on door-to-door delivery, usually gives 0·001in bore wear in 1,500-2,000 miles. The same vehicles with top rings chromium plated gave 6,600 to 13,000 miles per 0·001in bore wear.

Case B. A fleet of popular cars used

by travellers with normal rings gives 3,000-5,000 miles per 0·001in, but with plated top rings the mileage is over 15,000 per 0·001in.

Case C. A medium-priced car fitted with plated rings and wet liners and operating on long distance hard driving gave 38,000 miles per 0·001in.

Case D. A well-known commercial vehicle averaged about 3,000 miles per 0·001in with normal rings. With plated rings the minimum recorded figure is 12,000 miles per 0·001in and the maximum is over 40,000 miles.

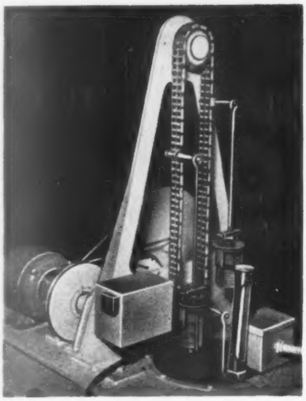
Comparative mileage figures per 0·001in wear for diesel engines are:—

	Miles
Hardened rings, regular oil	6,500
Hardened rings, detergent oil	12,600
Chrome rings, regular oil	15,000
Hardened rings changed to chrome rings at half mileage, detergent oil	20,700
Chrome rings, detergent oil	29,700

TESTING THERMOSTATS

TYPICAL of the care exercised by manufacturers to ascertain the reliability and effective working life of proprietary components fitted to their vehicles is the method now being used by the Research Division of Leyland Motors Ltd. to fatigue test cooling system thermostats. Such units are, of course, routine tested at the point of production but at Leyland they are being tested to destruction.

The laboratory-designed testing machine has a ¼ h.p. electric motor driving, through a series of speed-reducing belt drives, a shaft carrying a sprocket from which depends a short, endless, roller chain. From three laterally projecting rods, attached at equal intervals along the length of the chain, are suspended three thermostat units. In succession these are completely immersed in a tank of water maintained at a temperature of 195 deg F., raised clear of the water, and then again immersed.



Leyland machine for testing thermostat units

The speed of the machine is regulated to dip each thermostat every 72 sec. This provides a sufficient interval for the unit to cool before its subsequent immersion and ensures full operation of the valve gear. At the time of reporting, three thermostats produced by different manufacturers have completed 1,000 hr on test and are still functioning satisfactorily as the test is continued. It is difficult to relate the test regime to a mileage basis on a vehicle but the 50,000 immersions are estimated to be equivalent to twenty years operation on the road. When installed a thermostat seldom opens fully as the valve tends to maintain a constant water temperature.

An electric counter records the number of immersions and a separate meter shows the total running hours. To simulate adverse conditions under which a thermostat may have to operate, sand is added to the water in the tank. (2006)

BUS BODY-BUILDING METHODS

AT the Anglesey works of Saunders-Roe (Anglesey) Ltd., the building of bus bodies has been speeded by the use of a flow line assembly system. The employment of double-sided jigs for main components and the practice of pre-finishing units at the sub-assembly stage would seem to follow techniques developed in the aircraft industry.

Basic sections are the roof, sides and front and rear ends, all standardized. They are broken down into sub-assemblies and practically finished before erection on the chassis. All components are jig built and designed for knock-down methods of supply. The roof is built on a double-sided jig. It is skinned internally and externally, electrical components, handrails, bell cord and other details are fitted, and the inside is finished with cellulose paint. Side frames are built in pairs on double-sided jigs. Jigs are heavily constructed, and made with sliding locations so that any number of combinations of pillar pitches, heights and contours can be obtained with the minimum amount of setting time.

Chassis assembly

Preparation of the chassis is on flow line principles. Work flow is not continuous, but operations are balanced in time so that all buses in the line move together.

In the first stage the difficulty caused by flexibility in the chassis when working to fixed data was overcome by a drilling jig. This is a replica of the interchangeability points on the side, front and rear ends and is dropped into position and located on the chassis at pre-determined points. A water level system incorporated in the jig is used to compensate for the chassis flexibility and to strike a mean datum. Chassis brackets are then clamped to the jig, and out-riggers are drilled through into the chassis out-riggers from pre-drilled holes in the brackets. They are bolted into position and the actual pick-up holes for the body units are then drilled through from bushed holes in the chassis jig.

The chassis then rolls forward and the chassis bracket holes (which are now within the limits of interchangeability requirements) are used to locate supplementary jigs, which in turn locate the four wheel arch pans and the two step wells. One stage further on, the timber packing to carry floor boards is fitted and in the next stage the chassis is floored. Timber and other parts for these stages are pre-painted and made ready and brought to the line in kits.

Finally pre-cut lino is laid and the vehicle is driven into the main erection shop for the assembly of units.

The finishing line

The main units are taken from adjacent stacking jigs and are assembled on the chassis to form the completed body shell. Next the bus moves into a finishing line of four stages. Here joints between main units are covered externally by straps and internally by pressed metal finishers, sub-assembled access traps are fitted, doors hung, seats installed and the vehicle is then complete except for exterior painting. Painting is in five stages. Throughout the assembly lines the vehicle is pre-finished in sub-assembly detail, and unit stages, which cuts down the time a vehicle is using factory floor space. Interior painting is all done at the jig stage.

Planning ahead, the stores department arranges complete kits of details down to the last nuts and bolts.

In the detail fitting and sub-assembly shops, stillages with full kits of materials are brought to the worker by power or hand-operated lifting trucks, avoiding visits to the stores. (2010)

TRACTOR MANUFACTURE

A Survey of the Production Methods for the New Fordson Major

(*Continued from page* 67)

EACH type of engine that can be fitted in the Fordson Major has its own specific type of piston. Two machine lines are employed to produce the planned output. One line is used solely for machining pistons for diesel engines; the other is used for machining both vaporizing oil and petrol engine pistons. The petrol and vaporizing oil engine pistons have different dimensions but are similar in type. Both have a split skirt and two heat insulating slots at the top of the skirt across the thrust faces. Also on both, the skirt is a perfect ellipse at the top with the major axis across the thrust faces 0·014/0·016 in greater than the minor axis across the gudgeon pin faces. From the top, the skirt is tapered to the bottom to a circle 0·0015 in larger in diameter than the major axis of the elliptical top. Each of these pistons has a bottom scraper ring and a land scraper ring, but the vaporizing oil type has three and the petrol type two compression rings.

The piston for diesel engines has three compression rings and two scraper rings. It is circular and parallel from the bottom of the skirt to the centre-line of the gudgeon pin bosses, and has a solid skirt. From the centre line of the gudgeon pin bosses it is of barrel form to the bottom land. This form is such that the top of the skirt is an ellipse with the major axis 0·0115 in greater than the minor axis. The crown is machined to form a combustion chamber. Apart from this, the main machining operations are in principle the same for all three types of pistons. Therefore, it is only necessary to describe the methods used on vaporizing oil engine pistons to illustrate piston production.

Cast aluminium alloy pistons are used. At the first operation, which is carried out on a B.S.A. lathe, the skirt end is faced and the bottom of the skirt bore is machined to a depth of 0·28 in. Radial location for this operation is taken from two diametrically-opposed points on the inside wall of the skirt to ensure that when the O.D. is machined there will be even wall thickness. Length location is taken from the underside of the crown. A special internal chuck is mounted in the machine headstock. The casting is mounted on this with the open end to the headstock and the locators are brought into action by means of a hydraulic system. The same hydraulic system also moves the tailstock spindle forward to contact the crown of the piston to hold it securely in position. Machining is effected from both the back and front tool posts. The skirt end is faced and chamfered from the back while boring is effected from the front tool post. A cranked boring tool is used. Contrary to normal machine tool practice, the front tool slide feeds away from the headstock.

For the second operation the casting is transferred to a six-spindle Baird automatic. On this machine the O.D. of the skirt and the lands are finish turned, the crown is faced to length and all the grooves are finish machined. Location is taken from the faced end and the machined bore of the skirt. Loading and unloading are carried out at the first station.

At the second station on the Baird machine the crown is rough faced to length, all the ring grooves are finish machined to depth and a first turning operation is carried out on the O.D. The length of cut at this station is 5 in and the feed is 0·0065 in per revolution. The actual cutting time is 0·78 min. To facilitate setting up, magazine-mounted tools are used for machining the ring grooves.

Only two tools are used at the third station, one for semi-finishing the land diameter and the other for a second turning operation on the skirt. The length of cut is 3·50 in, the feed per revolution 0·005 in and the actual cutting time is 0·70 min. At the fourth station the crown is finish faced to length and an undercut is machined immediately below the groove for the land scraper ring. At a subsequent operation two heat insulating slots are milled in this undercut along the thrust faces. The length of cut at the fourth station is 2·12 in, the feed per revolution is 0·0045 in and the actual cutting time is 0·47 min. The land diameter is finish turned and the skirt diameter semi-finish turned at the fifth station, where the length of cut is 3½ in, the feed per revolution 0·0045 in and the actual cutting time 0·78 min. To complete the operations on the Baird machine, magazine mounted tools finish all the grooves to width and the skirt is finish turned at the sixth station at the same cutting conditions as are employed at the fifth station.

It will be noticed that four turning cuts are employed on the skirt, a sequence that has been adopted to allow very light cuts to be taken. This is important since it ensures that the cutting pressure is never sufficiently high to set up strains or distortion in the skirt. If strain were set up, it would be relieved when the skirt was split and the piston would lose its shape, and as a result difficulties would arise at the subsequent grinding operations on the skirt. A spindle speed of 1,000 r.p.m. is employed on the Baird machine. This gives a cutting speed of 1.040 f.p.m. at the second station. The complete cycle time is necessarily determined by the actual cutting times at the second, fifth and sixth stations, that is 0·78 min. The indexing time is 5 seconds, and the production at 80 per cent efficiency is 57 pistons per hour.

Several milling and drilling operations are then carried out. At these, oil holes are drilled in the ring grooves and the gudgeon pin bosses, the heat insulation slots

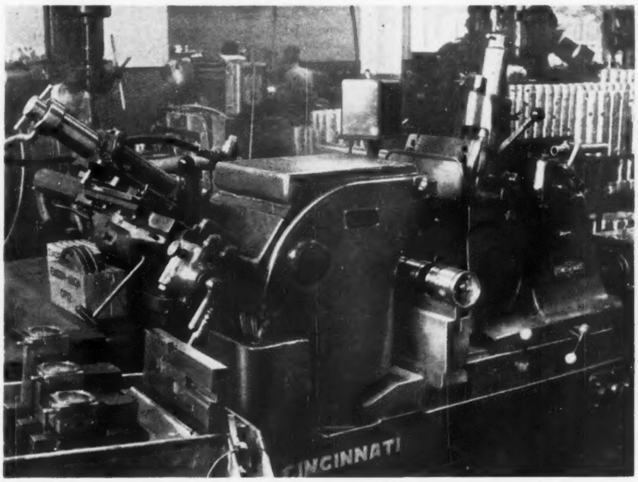


Fig. 26. Grinding the piston lands and skirt on a Cincinnati interrupted through-feed machine

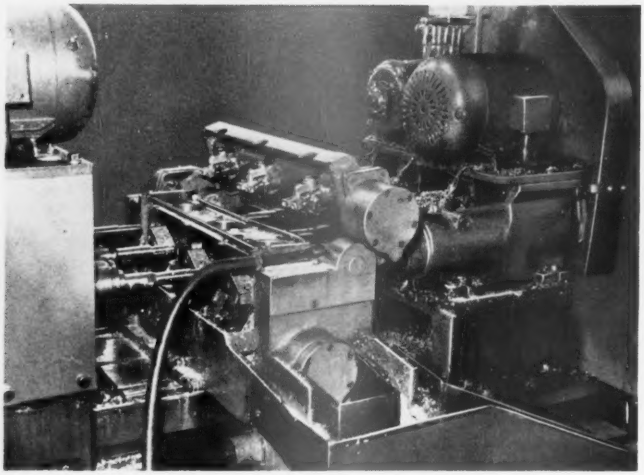


Fig. 27. The set-up for rough and fine boring the gudgeon pin holes on an Ex-Cell-O machine

and the vertical slot in the skirt are milled, and a clearance in the skirt end is milled.

The two following operations are of major importance. They are carried out on Cincinnati centreless machines arranged for plunge grinding with formed wheels. At the first of these operations both the land diameter and the skirt diameter are rough ground. In each case, to a tolerance of 0·001 in on diameter and leaving 0·007/0·008 in on diameter to be removed at the second centreless grinding operation. One of the machines is shown in Fig. 26.

These centreless grinding operations must be carefully controlled to avoid any marked degree of ovality. The form of the piston, particularly for petrol and vaporizing oil engines, is such as to call for very careful grinding if ovality at the bottom of the skirt is to be avoided. To begin with, across the gudgeon pin bosses the casting is much stiffer than across the thrust faces, with the result that the resistance to distortion under grinding wheel pressure varies round the periphery at the top of the skirt. At the bottom of the skirt the vertical slot reduces the resistance to slight collapse under grinding pressure.

Cincinnati through-feed interrupted plunge-cut machines are employed. The work is fed in down inclined rails with the control wheel in the out position. Correct endwise location in relation to the formed grinding wheel is obtained when the crown of the piston contacts a stop. The machine cycle is then started and the control wheel advances to bring the piston gently into contact with the grinding wheel. At the end of the pre-set advance of the control wheel there is a sizing dwell. When the pre-set dwell is completed a solenoid is activated to cause the stop to lift and the control wheel to retract to allow the ground piston to pass clear of the grinding wheel on to two sloping rails, down which it travels by gravity to a position convenient for the operating station of the next machine in the sequence. This centreless grinder can be operated either manually or on a fully-automatic cycle.

From the centreless grinding machine the piston is transferred to a double-end Ex-Cell-O fine boring machine, see Fig. 27. Each head of this machine carries three spindles. One head is for roughing and one for finishing, so that for every complete machine cycle the gudgeon pin bores on three pistons are fine bored to leave about 0·0002 in on diameter to be removed at a subsequent reaming operation. A work fixture with three pots for holding pistons is mounted on the machine table.

Loading is effected with the work fixture midway between the machine heads. Each piston is placed into a pot with the crown down and with the gudgeon pin holes approximately in line with the appropriate spindle. Through the operation of a hydraulic system, fingers on each side of the fixture are then brought into action to give accurate location. The head of the fixture, which is operated by the same hydraulic system as the locating fingers, is then lowered. Incorporated in the fixture head at each station are two arms which move out to contact the walls of the piston bore to ensure that the work does not move from the located position when the locating fingers are automatically withdrawn. This head clamps down on the end of each piston skirt.

There is a fully-automatic machining cycle. When it is initiated by push-button, the work table moves to the left and rough boring is carried out. At the end of the rough feed stroke, the table movement is reversed and the head travels to the right to the fine boring spindles. Incidentally, as little metal as possible is left for removal at fine boring. At the completion of the fine boring the table returns to mid-position for unloading.

An unusual feature of the fine boring operation is that there is no provision for clearing the tool from the work for the return stroke after fine boring is completed, yet the finished bore is completely free from return tool marks. A great deal of investigation into tool design was necessary to obtain this result. The cemented carbide tool is mounted in a split boring bar and is so shaped that on the return stroke the very small deflection of the spindle allows the back of the tool, and not the cutting edge, to contact with the machined bore. Surface finish from this

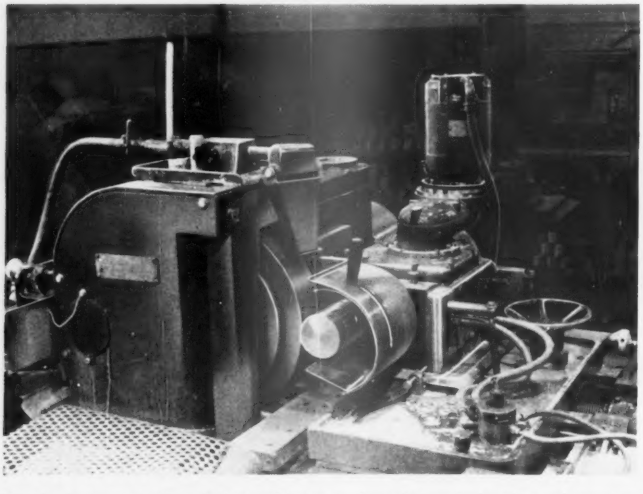


Fig. 28. Scrivener special grinding machine to produce the required ovality and taper on the piston skirt

machine is in the order of 15-18 micro-inches and the diametral tolerance is 0·0003 in.

After the gudgeon pin holes have been bored, the Circlip groove is machined with a conventional type of out-feed tool. The piston is then transferred to a special Scrivener piston grinding machine, see Figs. 28 and 29, on which the correct skirt form is produced. The ovality and the taper are producer simultaneously. Essentially, the general principle of this special grinder is that the piston is rotated about a horizontal axis and is at the same time given a controlled oscillation in a horizontal plane.

A normal Scrivener grinding head is used, but the machine incorporates a special work head. The driving spindle on which the piston is carried is mounted in a head that is arranged to pivot about a fixed point. A cam of appropriate shape is mounted on the spindle at the end remote from the work. The cam is maintained in contact with a fixed segmental follower by means of a spring. A two-lobe cam is used, each lobe coinciding with the axis of a gudgeon pin hole. The diameter ground immediately above the pivot point is a true circle, while the effect of the cam action increases as the distance from the pivot increases.

The pivoting workhead housing is carried on a main table which is driven by Scrivener controlled-cycle mechanism. It is the forward position of this table which determines the final dimensions of the work. The controlled-cycle mechanism is actuated by a cam which gives rapid advance, in-feed and rapid retraction in a completely automatic cycle. An ingenious guard is fitted to protect the operator's hands from the grinding wheel when he is loading and unloading the machine. It is shown in Fig. 28. When the table is retracted, the inner member of the guard is swung by the handle so that it is interposed between the grinding wheel and the workhead.

The final machining operation on the piston is interesting. It is the finishing operation on the gudgeon pin bores and is carried out on a Ford horizontal reaming machine with a special expanding reamer, see Fig. 30. The piston is mounted crown down in a pot-type holder that is mounted on the machine table, and is fully floating. The reamer is of special form and has a slight camber along the length of each cutting edge. Reciprocating motion towards and away from the machine headstock is imparted to the work through a hydraulic system. At the fine boring operation the holes are machined to 0·0002 in below the finished size. A few passes along the reamer brings the holes to size to a diametral tolerance of 0·0003 in and to a finish of 2 to 3 micro-inches r.m.s.

Although this description has not included any reference to inspection, there is very careful inspection throughout the machining sequence, particularly after the operations on the Baird multi-spindle automatic, the fine-boring and the grinding operations. There is a complete final inspection before the piston is tinned ready for transfer to the assembly department. Conventional types of stand gauges with several indicator dials are used to allow several dimensions to be checked simultaneously. This not only ensures that each dimension is within the specified limits, but also ensures that the correct relationships are maintained between the various dimensions.

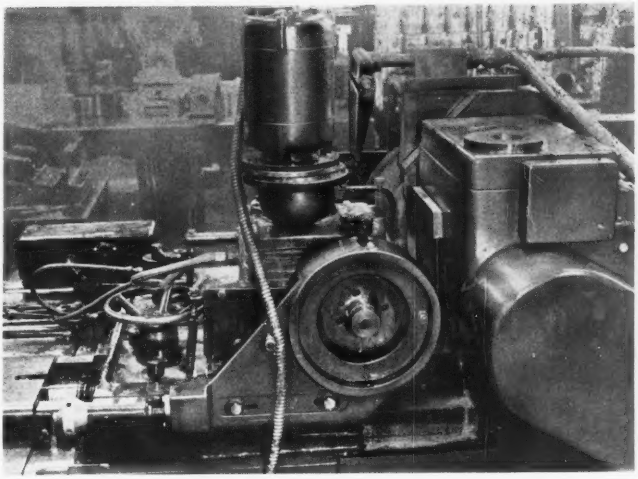


Fig. 29. The cam controlling the motion of the piston on the Scrivener machine

Transmission inner housing

Tractor manufacture entails the production of components that differ considerably from those that have to be machined in normal automobile practice. The methods used by Ford Motor Co. Ltd. for machining the front inner transmission housing may be used to illustrate the production of such components. This part, which is illustrated in Fig. 31, is a casting with an overall length of approximately 10¾ in, a body diameter in the order of 10½ in and with a flange at one end 12½ in diameter. Immediately below the flange is a spigot that is machined to 10·870 in nominal diameter. Two housings for shaft bearings are machined in each end of the body. Those in one end are finished to 4·3308/4·3317 in; in the other end, one is finished to 3·9368/3·9377 in and the other to 3·1494/3·1501 in diameter.

In addition to the close tolerances on separate elements of the component, relationships between one element and another must also be maintained within close limits. For example, the housing bores on the same axis must be

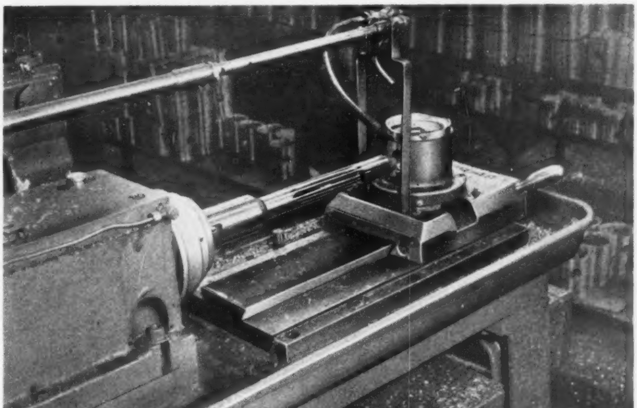


Fig. 30. Special Ford reaming machine for finishing gudgeon pin holes

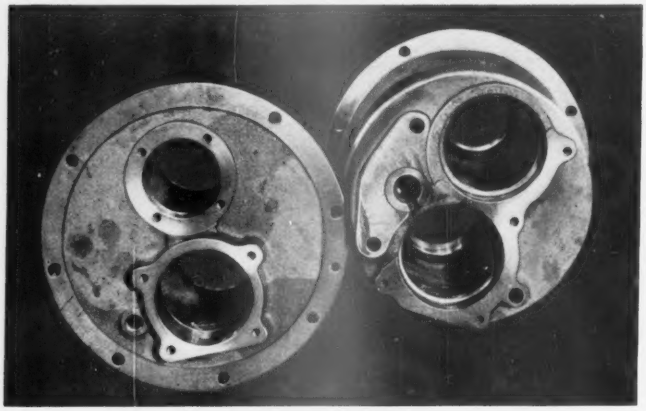


Fig. 31. Front and rear end of the inner transmission housing

Fig. 32. The set-up for the first machining operation on the inner transmission housing

Fig. 33. A Barnesdrill stub borer for machining four bearing housings

concentric within 0·002 in total indicator dial reading, as must the spigot diameter be concentric within the same tolerance with a turned diameter towards the other end of the body. In addition, the centre line of the spigot must lie in the same plane within 0·001 in as the plane joining the centres of the two housing bores in the flange end of the component. Furthermore, the back face of the flange must be square with the spigot within 0·002 in indicator reading.

For the first machining operation the casting is mounted in a pot type chuck on a Rydermatic machine, see Fig. 32. This chuck is in two parts, one arranged to grip the work immediately below the flange and the other to grip it towards the bottom. An interesting electric chuck key has been mounted on this machine to allow both chucks to be tightened simultaneously. The machine is tooled to rough and finish machine the outer diameter of the main flange and the flange face. A similar Rydermatic machine is used for the second operation. For this operation the work is located from the machined face and outer diameter of the flange. The work is clamped hydraulically through two cored holes in the flange end. On this machine the spigot below the flange is rough turned to a tolerance of 0·010 in to leave 0·030 in on the diameter for removal at a later operation. At the same setting, the inner face of the flange is machined and a diameter is turned on the body at the opposite end from the flange.

A double-end Barnesdrill stub boring machine, see Fig. 33, is used for the third operation. Each head carries two spindles, but only one at each end can be operated at a time, since the distance between the centres of the bores is too close to allow four spindles to be mounted for simultaneous operation. The work is located in the fixture for height from the turned diameter of the spigot, endwise from the inner face of the flange and radially from the idler shaft boss.

After the component has been loaded and clamped in the fixture, the machine cycle is fully automatic. To begin the cycle, the front pair of spindles advance simultaneously. The front boring bar on the left-hand head carries four boring and four facing blades to rough machine a bore to 3·130 in diameter and to face the spigot for this bore. This spindle runs at 200 r.p.m. to give a cutting speed of 165 f.p.m. for boring and 238 f.p.m. for facing, with a feed of 0·004 in per blade. Mounted on the front spindle of the right-hand head is a cutter that carries eight boring blades, in two banks of four, and one chamfering blade. The leading bank of boring blades finish machines a clearance hole at the bottom of the bearing housing and the rear bank rough machines the housing bore. This spindle also runs at 160 r.p.m. to give a cutting speed of 164 f.p.m. for finishing the clearance hole and of 180 f.p.m. for roughing the bearing housing bore. The feed is 0·005 in per blade.

As soon as the front boring heads have retracted, the work fixture is automatically shuttled to the rear to bring the other two holes into line with the rear spindles. These spindles also cycle simultaneously. The rear spindle in the right-hand head carries eight boring blades in two banks of four and a chamfering blade. The leading bank of boring blades finish machines a clearance hole for a shaft while the second bank rough machines the bearing housing. Cutting speeds are 170 f.p.m. and 180 f.p.m. respectively, and the feed is 0·005 in per blade.

There are four boring blades, four facing blades and a chamfering blade mounted in the cutter on the rear spindle of the left-hand head. These blades rough bore the bearing housing, face the spigot of the housing and chamfer the end of the bore. A cutting speed of 165 f.p.m. is employed for boring and of 250 f.p.m. for facing. The feed is 0·005 in per blade.

From the stub boring machine the component is passed to a third Rydermatic on which the elements rough machined at the second operation are finished to size. These elements are, the spigot diameter immediately below the main flange, the turned diameter at the other end of the body and the inner face of the flange. For the operations on the three Rydermatic machines the average cutting speed is 250 f.p.m. and feeds of 0·010 in per revolution and 0·80 in per minute are employed.

A special Cincinnati milling machine with two spindles at close centres is used at the next operation to mill the idler shaft boss and an adjacent inner face. To facilitate loading and unloading, the work fixture is mounted on rails. Location for this operation is taken from two turned diameters on the body for height, endwise from the inner face of the flange and radially from a rough machined bore. The cutting speed is 150 f.p.m. and the feed is 8 in per minute.

At this stage the component is transferred to a three-way drilling machine. This is a composite machine consisting of a Pollard double-end multi-spindle horizontal drilling machine with a single spindle vertical Pollard machine mounted at the rear. A cradle type work fixture is used in which height location is taken from the two turned diameters on the body of the casting, endwise location is taken from the inner face of the flange and radial location from one of the bores rough machined on the stub boring machine. For ease of loading the work fixture is mounted on bars which can be moved to bring the fixture clear of the machine. When the work is loaded and clamped into the fixture, a hydraulic system is actuated to move the fixture and the work to the drilling position. At the three succeeding operations, countersinking and tapping are carried out on the holes drilled on the three-way machine. For the drilling operations the cutting speed is 60/70 f.p.m. and the feed 3 in per minute. A tapping speed of 30 f.p.m. is employed.

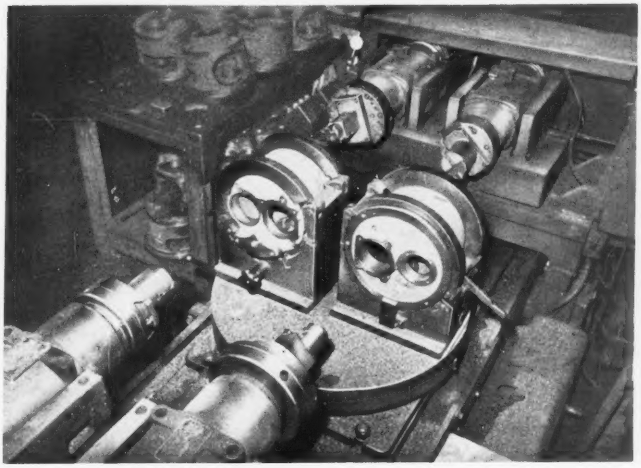


Fig. 34. Double-end Heald Borematic with special circular indexing table

The work is then transferred to a double-end Heald Borematic with two spindles in each head. This set-up is shown in Fig. 34. Two work-holding fixtures are mounted on a two-station circular indexing table. On this machine the main bearing housings are finish bored and faced, and a groove is machined in one of them. Height location is taken from the two turned diameters on the body, endwise location from the inner face of the flange and radial location from a reamed hole in the flange.

Loading is effected with the work fixture midway between the heads. A full automatic cycle is then initiated. The work moves to the left and one spindle bores a main bearing housing in the flange end of the component in the front fixture. At the same time the rear spindle in the left-hand head operates on the casting in the rear fixture and bores a bearing housing in the end remote from the flange. When the machining from the left-hand head is completed, the direction of table travel is automatically reversed and the work is carried to the right-hand head. From this head the front spindle is used to bore the bearing on the same axis as the bearing bored from the front left-hand spindle, and the rear right-hand spindle machines the bearing on the same axis as the bearing bored from the left-hand rear spindle. The table then returns to the mid-position and stops and the circular table is indexed through 180 deg.

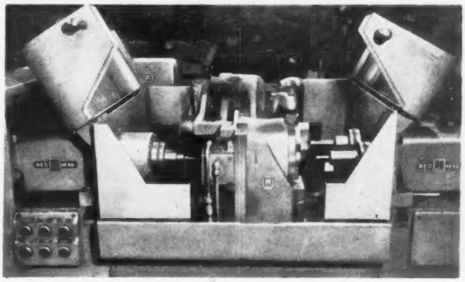


Fig. 35. The set-up for fine boring three holes in the inner transmission housing

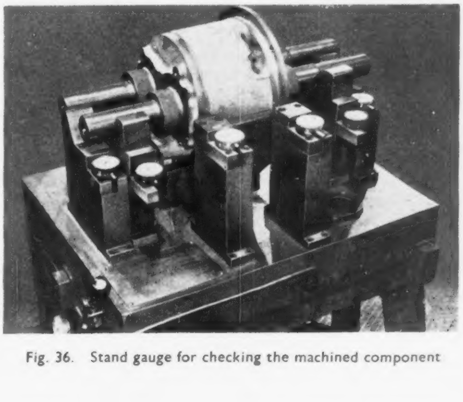


Fig. 36. Stand gauge for checking the machined component

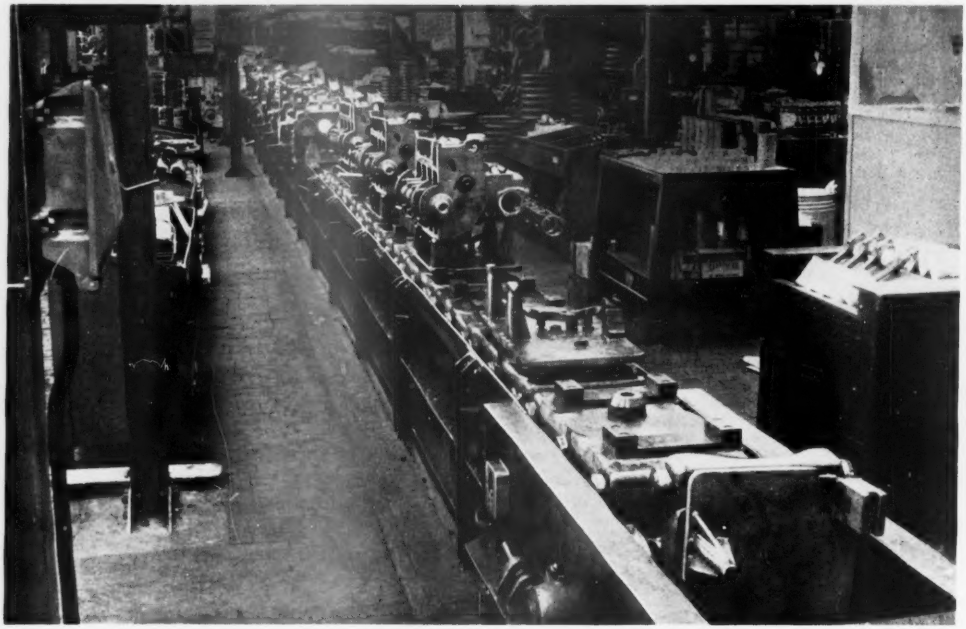


Fig. 37. Part of the common engine assembly conveyor for diesel, petrol and vaporizing oil engines

This brings a casting with all four holes finish bored into the unloading position. A cutting speed of 350 f.p.m. is employed and the feed is 0·007 in per revolution.

The major machining operations on the casting are completed at the next operation, which is carried out on a smaller double-end Heald Borematic with a single spindle in one head and two spindles in the other. This machine is shown in Fig. 35. On it the rear spindle of the right-hand head is used to finish bore and counterbore a hole in the idler shaft boss. The spindle speed is 880 r.p.m. to give a cutting speed of 256 f.p.m. for boring and 398 f.p.m. for counterboring with a feed of 0·005 in per revolution. At the same time the front spindle of the right-hand head is used for finish boring and counterboring one shifter shaft hole. The single spindle of the left-hand head is used for finish boring and counterboring a shifter shaft hole in line with the hole produced from the front spindle of the right-hand head. These spindles run at 2,000 r.p.m. to give cutting speeds of 322 f.p.m. for boring and 390 f.p.m. for counterboring, with a feed of 0·004 in per revolution. At the final operation, two holes drilled at an earlier operation in the idler shaft boss are reamed to size.

It is essential that not only must the various machined elements of this component be maintained to specified tolerances, but in addition the relationship spatially and as regards concentricity between the different elements must also be maintained to close limits relative to the size of the component. This result is obtained by using the same location points for all the major operations. Several stand gauges are used to check the work during the machining sequence. The gauge for the final inspection is shown in Fig. 36. It incorporates 10 dial indicators to give immediate and direct readings of the relationships between the various elements. An interesting feature of this gauge is that the cradle for the component is raised hydraulically for loading and is then lowered gently to the gauging position. The component with the two gauging shafts through the main bearings is weighty, and if it were attempted to place the work directly into the gauging position, there would be a possibility of affecting the setting

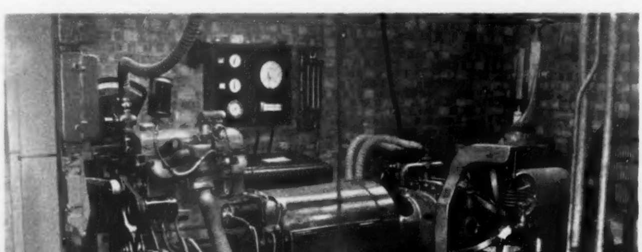


Fig. 38. Heenan and Froude test bed for Fordson Major diesel engines

of the dial indicators and also of damaging the gauging points.

Assembly

There are five main assembly conveyors, three for engines, one for transmissions and the final conveyor. The assembly department is rectangular with a gangway running longitudinally down each side. It is divided into two sections, separated by a transverse gangway. One section is concerned solely with engines. The other has on one side, spray booths for painting sheet metal parts and an assembly conveyor for transmissions. At the other side there is the final conveyor with the necessary assembly stations and painting plant. In the central portion of this section there is, towards one end, the equipment for assembling the tyres to the wheels. This equipment is conveniently placed in relation to the assembly station at which the road wheels are assembled to the tractor. Towards the other end of this central portion there are short sub-assembly lines feeding to the transmission conveyor.

Materials are conveyed to the assembly section by several methods. For example, most of the engine details are loaded direct on to an overhead mono-rail conveyor at the end of the machining section for direct transfer to the assembly department. Similarly, most of the transmission parts are conveyd to the assembly department on the same overhead mono-rail conveyor. This conveyor is so arranged that the parts are unloaded at positions convenient to the appropriate assembly station. Other parts are delivered in bulk by fork truck or other convenient means.

The lay-out of the assembly conveyors has in some degree been determined by the physical dimensions of the department. For example, one of the engine assembly conveyors is in two runs, one transverse and the other longitudinal. The transmission conveyor is also in two runs, first longitudinal and then transverse. In this case, however, there is the advantage that the transmission assembly finishes at a convenient place in relation to the end of one of the final engine assembly lines and to the start of the final assembly conveyor.

Engine assembly is started on a conveyor line that is common for all three engines, diesel, petrol and vaporizing oil. The cylinder blocks, which are common to all three engines, are taken off the mono-rail conveyor near to the start of the assembly conveyor, but before assembly starts the oil lines in the block are flushed out with kerosene and the block then passes through a hot washing machine. To begin the assembly, the cylinder block is mounted on a carrier plate with the sump face uppermost. Each carrier plate is mounted on four wheels running in angle sections. Movement of the conveyor is effected by a central chain on which the carrier plates are mounted.

At the end of the transverse run, the part-assembled engine is lifted by electric hoist and transferred to the longitudinal conveyor. This is a similar type of conveyor but it has different

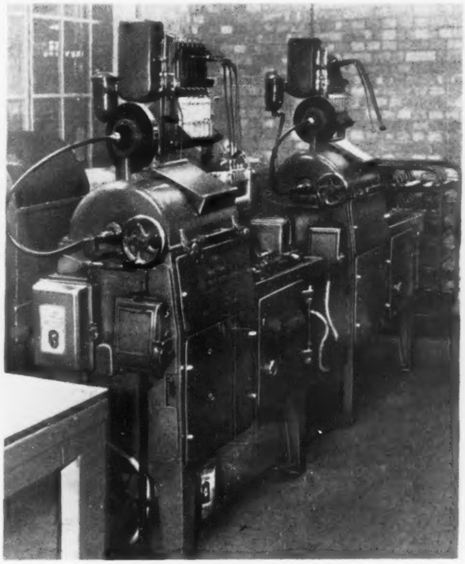


Fig. 39. Leslie Hartridge equipment for calibrating fuel injection pumps

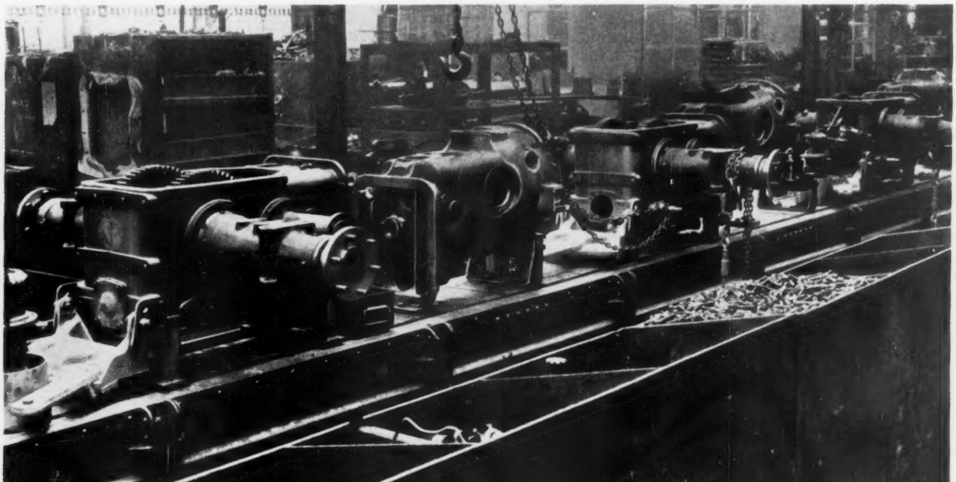


Fig. 40. Part of the transmission assembly conveyor

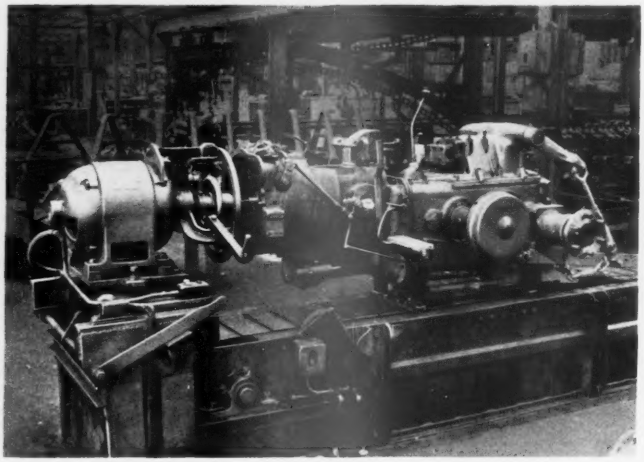


Fig. 41. A complete transmission assembly in position for a running test

carrier plates. In fact, two types of carrier plates are mounted alternately. At the beginning of this run, the block is mounted on carrier plates that can be indexed. It rests on the manifold face and through the first stations the rear end is leading. Work is carried out from both sides of the conveyor. This for example, allows the pistons and connecting rods to be inserted by one man from one side while another operator on the other side of the conveyor assembles the connecting rod caps and tightens them in position. At a certain stage along this conveyor it is necessary to work on the ends of the engine. At that point the carrier plate is indexed through 90 deg to bring the cylinder block across the line of conveyor travel with the sump face leading. Towards the end of this common engine assembly conveyor, the sump is bolted on. To allow this to be done, the engine is lifted off the carrier plate by electric hoist and is lowered on to the different type of plate immediately behind. Both types of carrier plates can be clearly seen in Fig. 37.

This common engine assembly conveyor runs between two floor conveyors on which final engine assembly is carried out. One of these conveyors is used only for diesel engines, the other is used for both petrol and vaporizing oil engines. Both floor conveyors have specially built cradles to bring the engine to a convenient working height, and each runs in the opposite direction to the common engine assembly conveyor. Transfer of the part-assembled engines to the appropriate conveyor is effected by means of an electric hoist on an overhead crane. Cylinder head sub-assemblies are produced on separate lines adjacent to the beginning of the relevant engine final assembly conveyor.

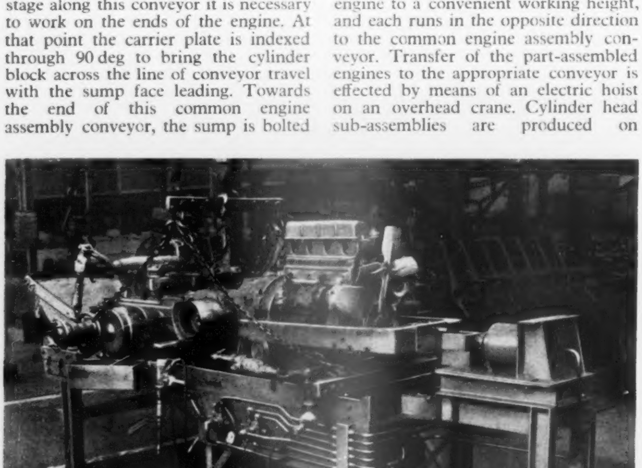


Fig 42. The rig for coupling the transmission to the engine

Diesel assembly

Assembly of a diesel engine, except for the fuel injection pump, is completed by the time it reaches the end of the final assembly conveyor. A calibrated pump is fitted to each engine during assembly, but is removed after the engine has been tested. At the end of the assembly line the diesel engine is loaded on to an overhead mono-rail conveyor for transfer to the engine test house which is located underground beneath the assembly department. Every engine is given a complete test on a Heenan and Froude hydraulic dynamometer. One of the test beds is shown in Fig. 38. Fuels pumps are all carefully calibrated on Leslie Hartridge equipment of the type shown in Fig. 39. From the test house, diesel engines are transferred by overhead mono-rail conveyor to the station in the assembly department at which the engine and transmission assemblies are coupled together.

With the necessary differences, petrol and vaporizing oil engines are assembled in a similar manner to diesel engines. It is not, however, considered necessary to carry out dynamometer tests on every one of these engines. Instead, every engine is run in for a definite period without being removed from the assembly conveyor. To allow this to be done there is a closed circuit floor conveyor that is synchronized with the assembly conveyor. Motors are mounted on the closed-circuit conveyor. At the appropriate point a motor is attached to an engine and at the end of the running-in it is released to be conveyed back ready for attachment to another engine. The engine assembly is completed a few stations beyond the running-in section. This brings the assembled engine to a position near the end of the transmission assembly conveyor and close to the start of the final assembly conveyor.

Transmission assembly

While engine assembly is proceeding, transmission assembly is carried out on another assembly conveyor, see Fig. 40. The initial stages are carried out on a slat conveyor running longitudinally towards the engine assembly section. A short secondary assembly conveyor runs parallel with the main assembly conveyor for transmissions. On it the front transmission housing is assembled. After the front assembly has been coupled to the rear assembly, the partly completed transmission is lifted by electric hoist and placed on another slat conveyor running transversely across the department. The end of this conveyor is adjacent to the end of the assembly line for petrol and vaporizing oil engines and to the start of the final assembly. At the end of this line, the transmission is coupled to a motor and given a short run, see Fig. 41. Incidentally, this and all the other assembly conveyors are automatically stopped when an assembly reaches the final

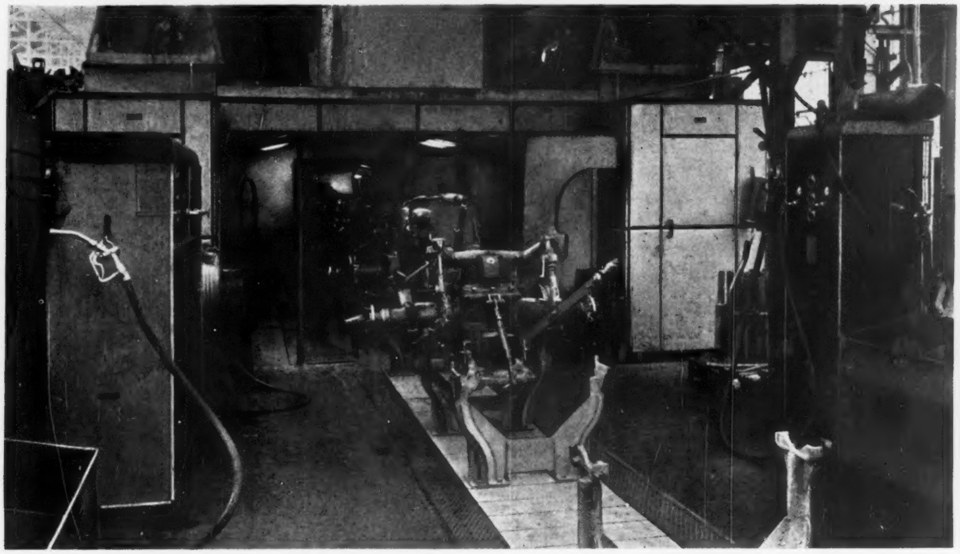


Fig. 43. The entrance to the paint spray booth on the final assembly conveyor

station and can not be restarted until the assembly has been removed. This is, of course, an essential precaution against inadvertent damage.

Hydraulically-operated equipment is used in coupling the engine to the transmission assembly. It is shown in Fig. 42. Transfer of engines and transmissions from the assembly conveyor to the coupling-up stand is effected by means of electric hoists on mono-rails. Similar means are used for transferring the assembly to the final assembly line which is of slat type at floor level with special cradles to carry the assembly at a convenient working height.

On its passage down this line, the assembly passes through a spray booth, see Fig. 43, then through a flash-off tunnel, through a second spray booth to a gas-fired infra-red drying oven, and finally through a cooling tunnel. For obvious reasons, certain components, chiefly electrical and road wheels, must not be exposed to heat. These are assembled to the unit after the painting sequence. Any necessary painting is then carried out in a third spray booth. The cradles on which the tractor is carried are of sufficient height to keep the wheels clear of the shop floor, but shortly before the end of the conveyor line, see Fig. 44, there is a small ramp on which the wheels make contact to lift the tractor clear of the cradle ready to be driven away under its own power.

Fig. 44. A completed Fordson Major leaving the final assembly conveyor

AUTOMATIC LOADER

Teleloader Device for End or Side Lifts

HYDRAULIC equipment to facilitate the loading of vehicles was recently introduced by Telehoist Ltd., Swindon Lane, Cheltenham. Known as the Teleloader, the device comprises a pivoted cantilever lifting arm actuated by a hydraulic jack.

Hydraulic power for operating the jack is provided by the standard pump used with hydraulic tipping gear made by the Company. The pump, operated by the engine running at idling speed, is mounted on the power take-off face of the vehicle gearbox. A control lever to engage the pump drive is fitted in the cab. The pressure pipe passes from the pump direct to the jack which is trunnion mounted between two steel brackets bolted under the floor of the body. In the return circuit, from the jack to the pump, is a hydraulic reservoir.

A simple, welded, cantilever, fabricated from steel tube, is employed as a lifting arm. At its root end, the upper members are carried on a steel tube mounted in tubular bosses welded to the jack support brackets. This tube acts as the fulcrum pin, and together with another shorter one, forming the piston rod attachment to the lower ends of the cantilever, it may be easily withdrawn to allow the lifting arm to be detached and stowed in the vehicle. The free end of the jack can then be swung on its trunnion mountings until a welded-on lug is aligned with two pendant stays welded one on each side of the mounting structure. The short piston rod attachment-pin is then inserted through the lug and pendant stays to secure the jack in its stowed position under the vehicle body.

Operation is fully automatic. When a sack is placed on the lifting arm, it pulls a webbing strap which opens the hydraulic control valve, and the lift commences immediately. A tension spring is included in the run of the control to keep it taut and to ensure smooth operation of the valve. The strap is connected to a roller chain where the control runs over the fulcrum tube of the lifting arm. This avoids trouble due to fraying of the webbing, and provides a relatively friction-free operation. Hand actuation of the valve may be resorted to should that be preferred. When the load has been raised to a convenient lifting height above the vehicle floor, operation ceases. On removal of the load the lifting arm is automatically lowered to the ground.

The outstanding feature of the Teleloader unit is its simplicity. It will lift as much as 3 cwt, although it might be considered as somewhat slow in operation. As the load is reversed during the lift the equipment can only be used to load closed sacks or containers. No doubt the unit has many uses in its present form and, with minor modifications, it could be adapted for many more. For instance, a parallel link motion could be used, in conjunction with a pivoted load-carrying end, to maintain open sacks in a vertical position during the lifting operation. Alternately, a telescopic end could be incorporated to assist in loading from a level higher than that of the ground. However worth while, such elaborations would, nevertheless, inevitably increase the present modest cost of the equipment.

The load, when resting on the strap, automatically operates the lift valve

In the lifted position, the bag is at shoulder height for ease of loading

CASTING OF STEEL IN ALUMINIUM DIES

IN comparison with permanent-mould castings made in the usual manner in cast-iron or steel dies, those produced from the aluminium-alloy dies employed in the Parlanti process are claimed to have an improved internal structure which is free from internal stress, localized chilling and cavities, as well as excellent surface finish and definition. This quality is achieved by two factors: (1) A high rate of cooling is possible owing to a high thermal conductivity of aluminium; it can be regulated by varying the surface area or the outside form of the die, or radiating heat from an inside source. (2) The provision of a large runner located at the thickest section of the casting ensures that thinnest sections are filled and solidify first, drawing heat from the hot metal behind them.

In an article in *Machinery* (*New York*), October, 1951, it is stated that the dies are usually made of the 2L33 aluminium alloy. Their surfaces are anodized to form a refractory barrier preventing fusion between the molten metal and the die, as well as to increase their wear resistance. In operation the dies are sprayed with a refractory coating such as sillimanite. Many simple methods can be used in the manufacture of the dies: direct machining for simple forms; casting the die parts against the anodized surface of an aluminium pattern; sand-casting in aluminium from a plaster pattern which has been cast from a wood master component or from a sample part. With large quantities a single pattern will serve for many dies which need only trimming, fitting and anodizing.

In addition to light alloys, iron and heat-resisting steels have now been cast satisfactorily by the process. When casting steel, an additional advantage is that alloying elements such as chromium or nickel may be eliminated, because the importance of their effects on the correct dispersal and formation of carbon is reduced by the rapid heat transfer involved in the process. Hollow castings may be produced with the aid of die-cast aluminium cores placed in the die. The core itself can be hollow and so made that it melts and flows out when the casting is formed. Applications are described in the article. (*M.I.R.A. Abstract No.* 5633.)

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Carburation. Volume Two

By Charles H. Fisher, M.I.Mech.E., M.S.A.E.

London: CHAPMAN & HALL LTD., 37, Essex St., W.C.2. 1952. $5\frac{1}{4} \times 8\frac{1}{2}$. 279 pp. Price 36s.

Originally published in 1939 under the title *Carburation and Carburettors*, this third edition has been divided into separate volumes. Volume One, reviewed in our March issue, dealt with the theoretical and technical aspects of carburation, whereas this volume deals with more of the practical side.

The work begins with a subject of great practical interest to automobile engineers and technically-minded motor car users. It should also be read by engine designers as well as advanced motor mechanics. This first chapter discusses the factors involved in fitting a new type of carburettor to an engine. In reading this discussion, one cannot fail to learn a great deal concerning the practical side of carburation, whether or not one is actually interested in fitting a new carburettor. The chapter includes much information on tuning and adjusting, including road tuning. Mileage tests are described and illustrated, as also are various jet calibrating machines.

The rest of the book is concerned with illustrated descriptions of all well-known makes of carburettor, each in turn. The Zenith carburettor being probably the most popular unit in this country is dealt with first, and several exploded views of differing types will be found of very great value to all mechanics and others who may need to dismantle these components. The author then passes to Zenith-Stromberg and Stromberg carburettors. The former is manufactured by Zenith and is standard on a number of British makes and the latter is widely used on American cars. Attention is directed to the differences between the two makes, especially as regards the main metering jets. This type of information, including gap settings at throttle edges, jet sizes standardized and special details of the Buick compound carburation system, methods of setting float levels, etc., make this a valuable chapter for the mechanic.

The horizontal and downdraught S.U. carburettors do not need so much space for their descriptions as they are simple to maintain and to describe. The various types of Solex are again well illustrated, both as regards the older types and the more modern ones incorporating economizers and accelerating pump systems.

A chapter is devoted to briefer descriptions of the Claudel-Hobson, Carter, Marvel - Schebler, Chandler - Groves, Holley, Ford and Tillotson carburettors and the Bristol TVO Tractor System and the Ford Vaporizer. The last chapter deals with automatic control systems for cold-starting and includes illustrated descriptions of the methods employed by Stromberg, Zenith-Stromberg, Solex (the "Thermostarter"), Carter (the Climatic Control), S.U., Chandler-Groves, Smith, Sisson and Delco-Remy.

An up-to-date book of this type is invaluable to automobile engineers. Carburettors can prove very troublesome when they are unbalanced or when it is required to overhaul them and tune them for specific conditions. This work cannot be undertaken without a comprehensive knowledge, and the information to be found in this book is not of the type that can be remembered in detail, reference to the illustrations at least would be essential.

Trader Handbook, 1952

London: TRADER PUBLISHING CO. LTD., Dorset House, Stamford St., S.E.1. 1952. $5\frac{1}{4} \times 8\frac{1}{2}$. 476 pp. Price 12s. 6d.

New material contained in this, the 46th edition of this handbook, includes the following items. Tune-up data is extended to cover further makes of foreign vehicles, although, incidentally, it is noted that Hotchkiss, Hudson, Jeep and Nash models are included in the British servicing list. 1951 models have been added to the Servicing Data table and specifications of 1952, up to date of publication, have also been included. Perhaps it would be as well to give the month of publication, since many models are frequently altering in detail. If the date is given, the reviewer has not been able to find it easily. Further additional matter in this issue is the write-up on servicing Sturmey-Archer gears for cycles and specifications of auxiliary engines for bicycles will be of interest to both garages and cycle dealers.

Other items have been brought up to date including the Legal Guide, the list of Trade Associations, Index of Registration Numbers (which give the dates when each number was used), local licensing offices, Traffic Area Headquarters, and vehicle licensing fees.

The book continues to provide a handy guide to all engaged in the motor and cycle trades, including both servicing and selling machines and component parts. The Buyers' Guide will tell at a glance who are the suppliers or manufacturers of every type of item required in these trades; such items as radio aerials, oil cans, caravans, carpets, defrosters, distilled water, gaskets, "L" plates, rims, solder, suppressors, white patches and yoke ends being a few representative examples selected at random from the long list. In fact all the day-to-day queries regarding who makes this or that, addresses of companies and official bodies, grades of lubrication oil recommended, sump capacities, etc., will most likely be solved readily by those who have this book to hand.

All-Purpose Diesels

By J. Malcolm Robson.

London: SIR ISAAC PITMAN & SONS, LTD., Parker Street, Kingsway, W.C.2. 1951. 6×9. 316 pp. Price 50s.

So many books are available on the design, maintenance, or the running of all types of diesel engines that it is difficult to find a new approach to the subject. The author states in his preface to this book that it is written in an attempt to present in a single volume an illustrated technical survey of the many problems attending the design and practical applications of diesel power units. He was formerly Chief Designer of the Turner Manufacturing Co., Ltd., and therefore has first hand experience of the design and running of small diesel engines. The book, however, covers engines of all sizes.

Early chapters devoted to engine types and applications, combustion processes, and fuel injection are followed by notes on component design. These notes provide an introduction to the more commonly used features of design as applied to crankshafts, flywheels, connecting-rods, valve gear, and pressure charging. The notes are more practical than theoretical and are obviously the result of long experience in the oil engine industry.

The chapter on Transmissions has been specially contributed by C. A. Pickering and covers clutches, couplings, belt drives, reduction gears, hydraulic couplings, dual drives, marine reverse gearboxes, marine stern gear, and anti-vibration engine mountings.

In his chapter on Starting, the author considers this under the three headings, factors affecting starting, aids to easy starting, and starting media. The latter includes hand, pneumatic, and electric methods. Three methods of governing are discussed, pneumatic, hydraulic, and centrifugal, and some hybrid types are also referred to. Cooling systems are illustrated and described, whilst the last chapter concerns itself with repair and maintenance. Some notes on lubrication and filtration are included, and a lengthy and complete "trouble-shooting" chart assists the engine user to find out why the engine does not start, or why it stops shortly after starting, or does not yield maxium output, or runs irregularly, or why exhaust smoke is black, etc.

A nicely produced book, easy to read and to understand, and one that will be appreciated by students of engine design as well as those engaged in the maintenance and running of diesel engines. Manufacturers of these engines will also find many among their staffs who would welcome a sight of this book.

Mechanical World Year Book, 1952

Manchester: EMMOTT & CO., LTD., 31, King Street West. $6\frac{1}{4} \times 4$. 268 + 360 pp. Price 3s. 6d.

The sixty-fifth edition of this popular handbook follows closely the lines of recent editions but, as usual, has been altered slightly and revised in places to bring it up to date. An interesting addition is an informative section specially written for this edition on the subject of patenting, since it is felt that encouragement of inventive genius will be of considerable assistance to this country during the present critical situation in regard to supplies.

The book is divided into two main sections. The first contains tables of all kinds including price equivalents, hydraulic data, weights and measures, brassfounders' metal mixtures, etc., to quote a few typical examples. The greater part of this section, however, consists of a Classified Buyers' Directory. The

second section opens with the new contribution on patenting, followed by notes on productivity and then going on to details of the continuous flow gas turbine. This second section includes many articles that are likely to be referred to by mechanical engineers generally, such as, for example, machine tools, die-casting, plastics, boilers, toothed gearing, steam turbines, etc. Most articles have been written by an expert on the respective subject. No doubt the handbook will retain its popularity.

Electroplating and the Engineer

By Alan Whittaker.

Manchester: EMMOTT & CO., LTD., 31, King Street West, W.3. 1951. 4¾ × 7¼. 87 pp. Price 4s. 0d.

The sixty-fourth booklet in the *Mechanical World* Monograph series provides an introduction to the techniques of electroplating. As the title implies, this book is intended for engineers rather than for electroplaters, although the latter may find some interest in it.

On the first page the author rightly stresses that high-quality plating cannot be expected unless high-quality materials are specified. The engineer who appreciates this, and after reading this book understands the implication of this statement, will benefit and his own electroplating plant will operate more efficiently as a result. Should he be ordering electroplating he will have a fundamental knowledge of the surface treatment required before plating and of the methods of plating, and this information will surely help him to avoid many pitfalls.

In the chapter on deposition data, plating methods using nickel, copper, cadmium, tin, lead, zinc and silver are briefly described before going on to chromium plating practice. The chapter on alloy plating includes speculum, brass, gold alloy and "bright" nickel plating. There is also a chapter on electrolytic metal finishing processes, and electrolytic polishing. Charts covering electrolytic metal finishing processes include the metals, their colours, characteristics and applications. They will be of value to the engineer and to anyone who uses or specifies electroplating.

Autocycles and Cyclemotors

By the Staff of "The Motor Cycle."

London: ILIFFE & SONS LTD., Dorset House, Stamford Street, S.E.1. 1952. 4¾ × 7. 140 pp. Price 5s.

The popularity of the motor-assisted cycle is revealed by the fact that tens of thousands of the compact "clip-on" motors have been sold in Britain during the past few years, and the slightly larger autocycles are equally in demand. Up to now there has been no really comprehensive and authoritative manual explaining these machines thoroughly, but this little book just published fills this gap most competently.

Users of these small machines are not as a rule very technically minded and frequently do not have the slightest idea how a machine works. The writers of this book have obviously kept this foremost in mind throughout. The working of the engine is described, together with fitting, adjustments, maintenance and repairs, and perhaps the most valuable and reassuring section from the novice's point of view is the detailed chapter on starting, running and stopping. Every type of cyclemotor available in this country is described, and every part of the mechanism of both autocycles and cyclemotors is illustrated and explained. More experienced owners will find many valuable hints throughout the book which even years of experience do not cover entirely.

Not only the mechanical point of view of these machines is dealt with, however, for there is information on legal matters such as licensing, and an extremely interesting section devoted to the cost of buying, running and maintaining motorized cycles, which will give prospective buyers real help.

The manual is written very enthusiastically, and even a brief glance through its pages would be enough to give an idea of the wealth of information and the simplicity of style which make it an encouragement and real help to the inexperienced user and a valuable reference book for the more mechanically-minded owner.

Hutchinson's Pocket Technical Encyclopædia

Compiled by L. E. C. Hughes, Ph.D., B.Sc. (Eng.), A.M.I.E.E., and Jean P. Bremner, B.Sc.

London: HUTCHINSON'S SCIENTIFIC & TECHNICAL PUBLICATIONS, 11 Stratford Place, W.1. 1952. 5⅛ × 7½. 182 pp. 7s. 6d.

The word "Technical" in the title of this book, the compilers explain in a foreword, is taken to mean "useful," and the words defined are those used by specialists in any field. A wide variety of subjects is covered, including Engineering, Physics, Chemistry, Music, Zoology, Botany, etc., etc., and there are also brief references to celebrated men connected with such subjects.

This pocket encyclopædia thus brings together a collection of words which the layman or a specialist in only one of the subjects treated would be unlikely to know the meaning. The definitions, of necessity in a small book, are very brief, but are more comprehensive than would be found in the usual dictionary of this size, since the words treated have been carefully chosen for their usefulness. Thus the advantage of this work is that it saves time being wasted in searching through large dictionaries or encyclopædias for the meaning of words used by people engaged in particular fields of science, or industry. It has been the aim of the compilers to present a quick summary of available information rather than to give a full exposition of each topic and, if that is borne in mind, this small book should be a welcome addition to general reference books, for the scope is wide and the information provided is up to date.

Motor Vehicle Technology Part 1

By R. W. Bent, M.I.Mech.E.

London: SIR ISAAC PITMAN & SONS, LTD., Parker Street, Kingsway, London, W.C.2. 1951. 4¾ × 7¼. 104 pp. Price 7s. 6d.

In spite of the wealth of books dealing with automobile maintenance and repair, there are few books of value available at low cost to suit the pocket of students and young mechanics who wish to supplement their practical knowledge with easy-to-understand technical details. This book has been specially prepared for those students working for the examinations of the City and Guilds of London Institute (Motor Vehicle Service Mechanics), the Union of Lancashire and Cheshire Institutes, the East Midland Educational Union, the Union of Educational Institutions, and the Institute of the Motor Industry, fills this need. In fact, it has filled such a need since the first edition in 1946.

This second edition is similar to the first except that more up-to-date illustrations have been substituted where the former ones did not represent current practice. These illustrations are clear line-drawings of definite value to the student. The book will probably be read mostly by those who have had little practical experience, but it will also be useful to those who are engaged in maintenance work. It covers such subjects as engine details, lubrication, cooling, carburation, fuel supply, air filters, ignition, and transmission. Each chapter concludes with a number of questions that have appeared in examination papers set by the various bodies mentioned above.

The Welding, Brazing and Soldering of Copper and Its Alloys

Radlett, Herts: THE COPPER DEVELOPMENT ASSOCIATION, Kendals Hall. 1951. 5¾ × 8½. 188 pp.

The Copper Development Association are to be congratulated on producing this valuable book on the jointing of copper and its alloys and presenting it free to those to whom it will be of service. We imagine that there will be a great many engineers who will find the book a mine of information. Commencing with metallurgical and physical properties, it passes on to main welding methods which include oxy-acetylene welding, arc welding and resistance welding. Chapters are devoted respectively to the welding of copper, of copper alloys, and bronze. The alloys include those with high copper content, brasses, gunmetals, aluminium bronzes, silcon bronzes and copper-nickel alloys.

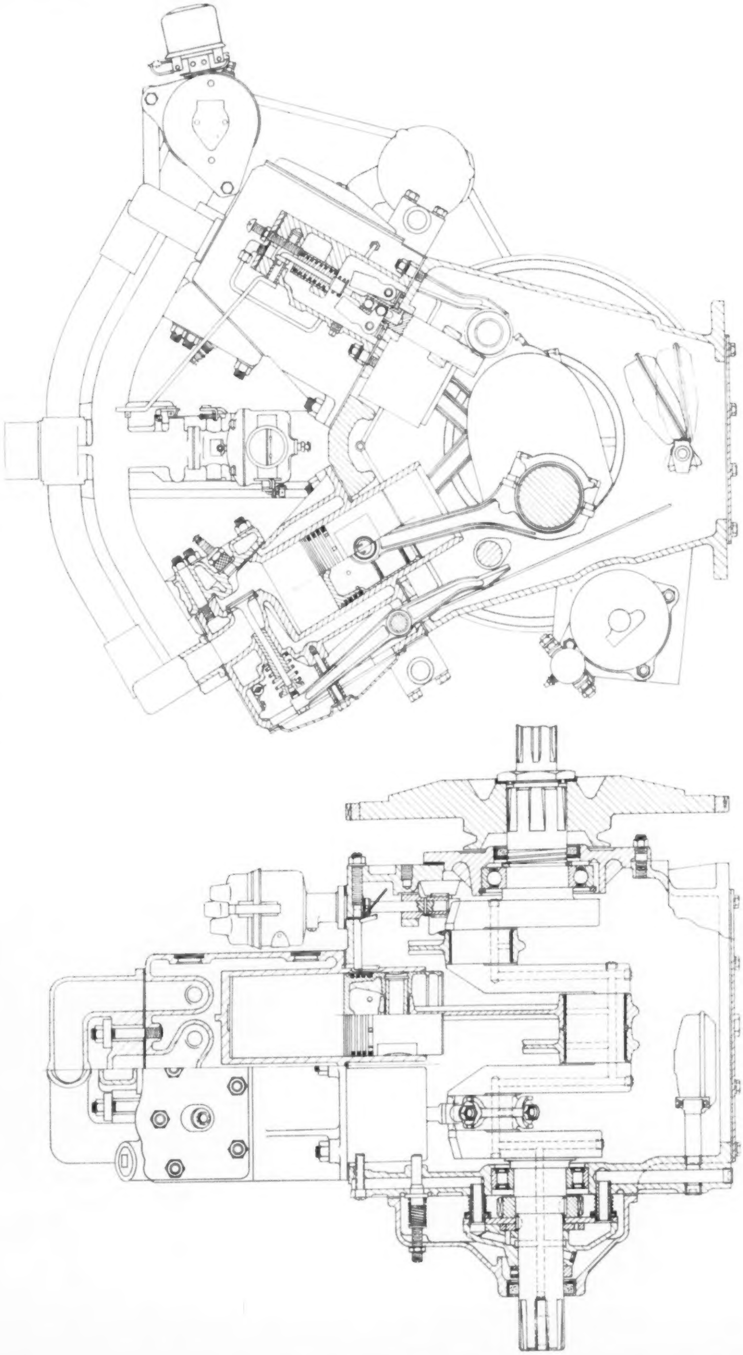
Under Brazing and Silver Soldering are included some notes on copper as a brazing material for steel, a subject of growing importance in modern production methods. Other chapters cover the soft soldering of copper and its alloys, additional jointing and related processes (including atomic hydrogen welding and submerged melt welding), and finally, the jointing of dissimilar metals.

The book is profusely illustrated with both half-tones and line drawings and includes a number of useful tables and charts, and also a valuable bibliography. This is the 47th book issued by this Association. Readers should give evidence of their responsible status when applying for a copy.

Mechanical World Electrical Year Book, 1952

Manchester: EMMOTT & CO., LTD., 31, King Street West. 6¼ × 4. 360 pp. Price 3s.

In its forty-fifth year of publication, this handbook still continues to provide considerable value for money. The section on electric motors has been completely rewritten and is now technically up to date. Similarly "Dynamo and Motor Defects" has been revised. Otherwise, this book remains as before, and provides a handy reference book not only for electrical engineers, but for mechanical engineers also. Some of its chapter headings include Electrical Calculations, Turbo-alternators, Electric Braking, Transmission and Distribution, Control Gear, Rectifiers, Meters, Electric Welding. Many tables, charts and graphs are included as usual.



GENERAL ARRANGEMENT OF MINNEAPOLIS-MOLINE V-4 ENGINE

Bore and stroke $3\frac{5}{8}$ in × 5 in
Swept volume 206·41 in^3

MINNEAPOLIS-MOLINE ENGINE

A Four-cylinder, V-Type Unit for Self-propelled Farm Machines

IN recent years self-propelled, as distinct from tractor-drawn, agricultural machines have become increasingly popular in America and are now receiving some attention in Europe. The obvious example is the combine harvester but it would seem logical to apply the principle to other farm machines such as beet, potato or forage harvesters; ploughs; drills; mowers; windrowers; and hay balers. Self-propelled machines are highly manoeuvrable, convenient to use and, in most instances, effect a saving in labour as only a single operator is needed. These advantages, however, are outweighed by the higher aggregate capital investment required and the fact that, due to their specialized character, such machines are operated for relatively short periods each season. As a consideration of general economy, they lead to the multiplication of power units and other components.

To circumvent these disadvantages, resort is made to a propelled " universal carrier " on which the various specialized machines can be interchangeably mounted. The attachment is designed to ensure a minimum expenditure of time for a changeover of machine units. To remove a machine, the carrier is driven under a simple frame hoist, a sling fitted, the driving belt removed, two pins of the hydraulic implement lift gear withdrawn, and the three attachment points disconnected. When, by means of a hand-cranked hoist, the machine is raised clear, the carrier is backed away and the machine lowered on to a simple dolly for transport and storage.

A German carrier, the Lang, is a four-wheeler steering on all wheels but the Minneapolis-Moline is a three-wheeler (two-tracked) and steers on the single rear wheel. An experimental carrier was tested in 1946 and subsequently put into production.

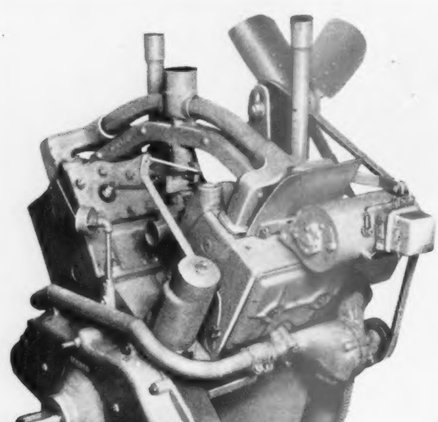
Design considerations

An essential component was a simple, compact and robust power unit and for this an interesting design was evolved. Special consideration was given to limitations of space, weight, character of drives, minimum spares and low production cost. To reduce tooling and manufacturing costs a substantial percentage of the component parts were to be common with those for the engine of the Company's Model Z tractor engine which has been in production for a number of years.

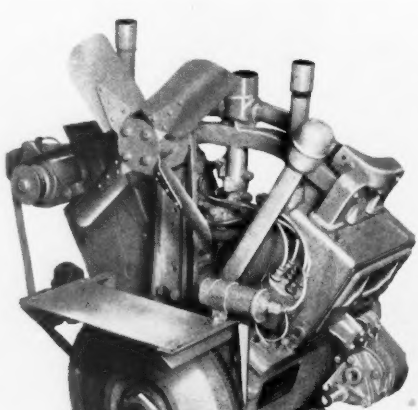
For compactness, a four-cylinder, V-type design having a three-throw crankshaft is used. As compared to the conventional 90 deg V-type engine, the choice of a 60 deg cylinder block angle gives a reduced overall width; less vibration, it is claimed; and a smoother firing order of 120–180–240–180 deg instead of 90–180–270–180 deg. The bore is $3\frac{5}{8}$ in and stroke 5 in, giving a total swept volume of 206·41 in^3 (3382·27 cm^3). Compression ratio is 6·15 : 1 and at 1,500 r.p.m. the output is 38 b.h.p.

Cylinders, incorporating the heads carrying the horizontal valves, are cast in pairs and are separate from the barrel-type crankcase. On each block a shallow detachable cover, bolted to a facing on the inner side and arranged at an angle of 15 deg to the plane containing the cylinder axes, closes the combustion spaces and furnishes a means of access to the valves.

A drive is taken from each end of the crankshaft and, to sustain the belt loads, large diameter anti-friction bearings are used for the shaft mains. The shaft is threaded into the crankcase from the rear and supported in a roller bearing in the front end wall and a ball bearing in a housing registered in the rear wall. Crankpins are



Left-front view



Right-rear view

Minneapolis-Moline engine for universal carrier

2·62 in diameter and 1·28 in long, the centre one being of double length to accommodate a pair of rods side-by-side.

Forged steel connecting rods have a centre-to-centre length of 8¾ in and a diagonally divided big-end with the cap secured by two set bolts. Complete with bushing each rod weighs 45 oz. Cast iron is used for the flat-topped piston, which is fitted with three compression rings and one oil-scraper ring. All rings are above the 1·00 in diameter, 3·00 in long hollow gudgeon pin retained by circlips. With rings and pin each piston weighs 54 oz.

Valve gear

Interchangeable inlet and exhaust valves have 1·46 in diameter heads with 45 deg faces, 0·341 in diameter stems, and lift is 0·355 in. Alloy cast iron seating inserts are provided for exhaust valves only. As the valves are actuated directly by long rockers mounted outside the cylinder blocks, two camshafts are necessitated. These are mounted in the crankcase and driven from a helical gear keyed to the front end of the crankshaft.

Lubricating oil is carried in the crankcase sump and a novel feature of the system is the floating oil pump on the front of the crankshaft, immediately forward of the helical timing gear. This enables oil to be circulated to the big-end bearings by way of the drilled shaft, despite the use of anti-friction main bearings.

Auxiliary drives

Just forward of the left hand cylinder block, the fly-ball governor, and just to the rear of the right hand block, the distributor, are each driven, by spiral gears from their respective camshafts. Other auxiliary equipment is driven at the rear by V-belt from a pulley formed on the hub of the flywheel. This includes the radiator fan, the dynamo carried on the left hand cylinder head, and the water pump bolted to the outer side of the left hand cylinder block.

The dynamo, it will be noted, is protected by a shield from heat radiated by the overhead exhaust manifold. A starter motor mounted low on the right of the crankcase engages a ring gear shrunk on the rim of the flywheel. Mixture is supplied by a Marvel-Schebler carburettor.

Transmission

The transmission is somewhat unconventional. One half of the driving sheave is formed by the flywheel; the other half is mechanically operated to open or close the belt groove in conjunction with complementary movement of the inner half of the driven sheave on a countershaft. An idler contacts the V-belt and a slight pressure is sufficient to take up slack. The gearbox provides three forward speeds and a reverse and, in conjunction with the variable belt drive make it possible to operate the carrier at any speed from 1 m.p.h. to 10 m.p.h. The calculated speeds are:

Low	0·96 to 2·20 m.p.h.
Second	1·90 to 4·50 m.p.h.
High	4·30 to 9·78 m.p.h.
Reverse....	0·66 to 1·90 m.p.h.

The final drive is by a 3 in heavy duty, endless belt which also serves as a clutch by means of a spring idler which can be released by manipulation of a pedal. This clutching system has been used by the firm for a number of years on the cylinder drives of combine harvesters.

Overall dimensions of the engine, which is mounted transversely on the carrier, are 33 in wide, 34 in high and 31½ in long.

MORAINE-400 ENGINE BEARINGS

IN a *S.A.E. Preprint*, January 1952, A. R. Shaw describes the development of Moraine-400 bearing material. A cadmium-silicon-aluminium alloy, GM-3889-M, with excellent ductility and score-resistant properties, was taken as the starting point. Because of the difficulty of bonding aluminium to steel, previous aluminium bearings had been solid in structure, but the relatively high coefficient of expansion of aluminium made such bearings unsatisfactory. On the other hand, the relatively heavy layer of brittle iron-aluminium alloy produced at the bond by existing processes of bonding aluminium to steel gave the composite material inadequate fatigue resistance. Moreover, these processes were unsuitable for large-scale bearing production. Eventually, however, a process was developed which gave the strong ductile bond required in engine bearings. In dynamometer tests, the plain aluminium-clad steel bearings generally ran well, but occasional failures indicated the necessity of a babbitt overlay to provide anti-scoring properties.

The method adopted was that of electro-co-deposition, but there was no experience to draw upon for application to aluminium, and a satisfactory process had to be worked out. After eight years of development and three years of dynamometer testing, the composite bearing material Moraine-400 has been evolved. It consists of a steel backing clad with about 0·010 in of the cadmium - silicon - aluminium alloy which is itself overlaid with a tin-copper high-lead babbitt by electro-co-deposition to a thickness of up to 0·010 in; the aluminium is now merely one of the components. Moraine-400 has from six to ten times the bearing life of the conventional babbitt which, in a particular new engine, withstands about 100 hr of endurance testing, while the Moraine-400 bearing runs 750 hr with ease. Their general adaptability and conformability allow Moraine bearings to give satisfactory operation after being changed from one engine to another. To date there have been no failures attributable to fatigue or bond failure. Where bearings have been properly designed and installed, there have been no failures from scoring. The embeddability is "phenomenal," and the aluminium and babbitt ensure high corrosion resistance. (*M.I.R.A. Abstract No.* 5746.)

Daimler Regency Gearbox

IN the article on the Daimler Regency Gearbox, published in the March 1952 issue of the *Automobile Engineer*, the linings for the brake bands were specified as "Halo," supplied by Ferodo Ltd. However, "Mintex" is the material used, and it is manufactured by British Belting and Asbestos Ltd., who state that "Halo" has been a registered trade name of one of their products since 1922. This name was applied to a specific class of synthetic resin-bonded wound yarn friction linings. Considerable quantities have been produced over the last thirty years for a wide variety of purposes, including substantial deliveries for various forms of Wilson pre-selective gearbox bands. Since 1945, the use of the word "Halo" has been abandoned by the Company.

The general brake lining name of "Mintex" is now applied to these wound types, as well as to the more normal woven and moulded varieties of lining produced by their company.

A wire-woven Ferodo lining is used on Daimler armoured car and tractor gearboxes of the Wilson type.

REDUCING ROAD NOISE

A Silentbloc Frustacon Unit Applied to Coil Spring Suspension Systems

MUCH attention has been devoted, of recent years, to the reduction of noise in motor cars. The most difficult part of the problem is to eliminate the noise generated by the contact of the tyres with the road. This noise is particularly noticeable when travelling over a freshly made road surface as, under these conditions, the surface is usually rougher than one that has been in use for a year or more. Engineers and physicists, who have carried out investigations on this subject, are not all in complete agreement as to why road noise seems to be more apparent in modern cars than in earlier ones. However, there are probably several contributory factors.

One of these is that engine mountings have been steadily improved, and now, as a result, hardly any vibration is transmitted to the structure. In fact, when measured with a noise level meter, there is little or no difference between the noise in a car travelling with the engine running, and that in the same vehicle moving at the same speed with it switched off and out of gear. It follows that the engine noise which used to mask effectively the other noises, has now been reduced to the stage where it is the other noises that are the more noticeable.

Another feature of the modern car is the enveloping wings and body. In many cases, the body is constructed integrally with the chassis frame, and it can even be said, of some cars, that there is no chassis frame. Moreover, welding is now employed extensively in places where bolting was formerly used, with the result that the damping characteristics of the structure are greatly reduced.

A change that has often been blamed for increasing the noise transmitted to the structure is the incorporation of independent front suspension. With this system, two factors are considered to have had a detrimental effect. One is that the older leaf springs had,

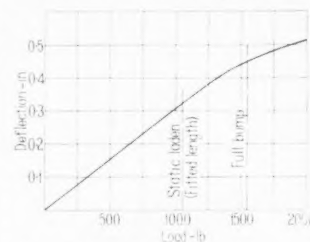


Fig. 3. Typical load deflection curve of a Frustacon mounting

owing to interleaf friction, a considerable inherent damping capacity, whereas the coil spring or torsion bar has not. The other factor is that the transverse links, of the independent suspension system, directly connect the unsprung mass with the chassis.

The logical, and probably the most economic measure to adopt, for the reduction of the transmission of road vibration to the vehicle, is to insert rubber between the suspension and its mounting brackets. Rubber bushes are employed for this purpose in the pivot joints of suspension units. This has been done by many manufacturers, but the effectiveness of the measure is limited by the fact that the steering geometry must be accurately controlled. The employment of rubber bushes in these joints has, of course, an additional advantage in that there is no need for lubrication.

However, the noise problem is far from being completely solved. Accordingly, attention has been given to the mounting of the spring, since, in many designs, this is the only path remaining along which vibrations can be transmitted through a metal-to-metal joint. Apart from the direct transmission, from the coil spring to the chassis, of road excited vibrations of the unsprung mass, there is evidence to indicate that surging of the spring takes place. This can be observed when a chassis is placed with its wheels over rotating drums fitted with bumps or an artificial road surface. Under these conditions, by placing the hand on the coils of the spring, the vibration can also be felt.

The surge takes the form of a vibration at one of the natural frequencies of the spring itself. Suspension springs have a relatively high mass, therefore the energy of the vibrating system is fairly large. Accordingly, it is reasonable to suppose that the end reactions to the surge will be appreciable, and that the vibrating spring is a potential source of structure-borne noise in vehicles.

The tyre is capable of transmitting the fairly high frequency vibrations that may excite the spring surge and that may also be communicated directly through the spring to the chassis frame and body. Despite the fact that the best way to reduce noise

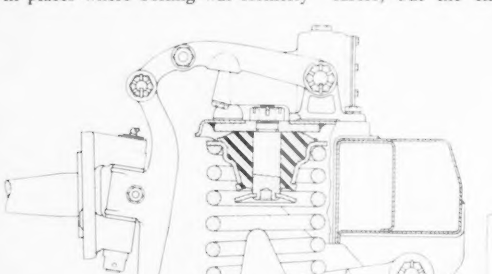


Fig. 1. An application of a Silentbloc Frustacon mounting to an I.F.S. System

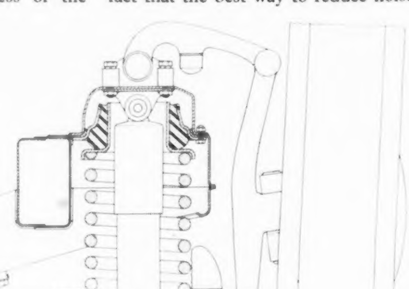


Fig. 2. There are alternative designs to suit different suspension layouts

is to stop, at the source, the vibration producing it, in this case there seems at present to be no way of doing so. Although the tyre is intended to absorb the high frequency excitation caused by small road irregularities, it cannot perform this function entirely satisfactorily. This is because adjacent to its side walls, it is relatively stiff so far as loading in the vertical plane is concerned.

An alternative method of reducing noise, by isolating the chassis from the spring has been tried. Hitherto it has, in some cases, been the practice to insert a rubber washer, about 1 in thick, between each end of the spring and its seating. Although this has, no doubt, a beneficial effect, due to damping, it can be improved upon. Like many improvements, it can only be effected at an additional cost, but at least one manufacturer has decided that this is warranted by the results obtained.

Incorporated in the front suspension of the latest Rover 75 is a Silentbloc Frustacon anti-vibration mounting. The unit consists of three components; an outer steel pressing formed with an external circular flange which replaces the normal coil spring lower pan of the Rover 75, a steel inner sleeve, and a cone shaped rubber insert which separates the two steel members from each other. The unit is secured to the lower wishbone member by means of a bolt through the centre of the inner sleeve. Mounted on the upper end of this bolt is a dished washer which forms an abutment for the bump stop fitted to the coil spring upper pan. Thus the bump load is not taken entirely through the rubber of the Frustacon unit.

Other layouts are practicable and the unit may be fitted at either the top or bottom of the coil spring. Fig. 1 shows the mounting in the top position. In this case, the unit is mounted on a central spindle, the top end of which is shouldered to fit a hole drilled in the upper portion of the suspension bracket, to which it is secured by a slotted nut.

A bump stop is incorporated in the unit by securing a heavy gauge disc on to the lower end of the central spindle. The rubber stop is mounted on the lower spring support pan and, in the full bump position, it bears against the disc and spindle. It is usually regarded as fundamentally sound for the spring and bump loads to act along the same axis, as a common supporting structure can then be used for both functions. This leads to simplicity and efficiency, reducing the weight to a minimum.

Apart from limitations imposed by the need to keep the price as low as possible, several other factors have influenced the design. In order to keep down the overall height of the suspension unit, and thus to reduce to a minimum the weight of the supporting structure on the chassis frame, the vertical dimensions of the rubber mounting must be restricted. To satisfy these conditions, a flange at the top end of the outer member carries the spring. With this arrangement, it is possible to incorporate the amount of rubber necessary to carry the load without being unduly restricted by space considerations.

Another feature desirable for this type of mounting is that under normal loads, its rate shall be as low as possible. This characteristic has been incorporated by giving the unit a variable rate, relatively low to support normal loads, but increasing with deflection. The load-deflection curve for one of these mountings is given in Fig. 2, but the manufacturers state that it may be altered to suit particular requirements of individual designs, without changing the physical dimensions of the unit. The rubber is not highly stressed by the heavier loads as, under these conditions, the outer member and the upper rubber flange bear against the support bracket so that the main portion of the rubber member is relieved of excessive loads that might, in time, damage it.

A Silentbloc unit designed for use in vehicles employing a telescopic shock absorber co-axial with the coil spring is illustrated in Fig. 3. With this arrangement, it is usually the shock absorber that governs the overall height of the suspension system and, in some designs, it will project through the upper support bracket. For this reason, the inner member of the Silentbloc unit is a tube of a diameter sufficient to clear the shock absorber, the angular movement of which is catered for. The spring seats on the flanged lower end of the inner member, and the outer member is bolted to the suspension bracket on the chassis frame. No bump stop is incorporated, but otherwise the characteristics of the unit are similar to those of the Frustacon mounting. Some shock absorbers have built-in bump and rebound stops.

TRANSPARENT CUTTING OILS

FOR a long time now cutting oils have contained various additives, such as sulphur or chlorine, which, while improving the cutting characteristics, tended to darken the oil. Methods have now been developed, states *Lubrication*, January 1952, permitting the application of considerable quantities of additives without affecting the transparency of the oil. The main advantage derived from the use of transparent oil is that the tool and the workpiece can be observed during the machining, and thus improved accuracy and finish are possible. Inspection of the parts produced can more easily be accomplished without stopping the machine and removing the oil. In addition, there is a psychological effect on the workers, resulting in a higher efficiency of work due to cleaner operating conditions.

Cutting oils designed for heavy machining operations contain sulphur in active form, and such oils will corrode copper at room temperature. In other respects transparent cutting oils of this type are outstanding: they dissipate heat rapidly, provide efficient operation, carry-off is minimized, and they readily separate from chips and contaminants. In cutting performance, they are far superior to transparent oils available not long ago, although sometimes they do not equal the best black oils. A non-corrosive type of transparent cutting oil has also been developed which, although less effective than the active type, is suitable for machining copper and its alloys. Exceptional properties of some non-corrosive sulphurized oils permit their application as both a cutting fluid and machine oil, and also as the hydraulic medium if the machine is so equipped; the danger of mutual contamination of the different oils is thus eliminated.

A chapter on dermatitis discusses the action of cutting oils on the skin, bacteria in oils, and methods of prevention and treatment of skin infection. (*M.I.R.A. Abstract No.* 5735.)

Ultrasonic Flaw Detector

THE field for non-destructive testing is large and ultrasonic apparatus, in conjunction with an oscillograph, is being widely used for the examination of castings, forgings, machined parts and welded joints. As the scanning trace indicates an initial pulse at the entry surface and a bottom echo from the remote surface, intermediate peaks caused by discontinuities, inclusions, or porosity can be readily located. A new portable flaw detector manufactured by Solus-Schall Ltd., 18 New Cavendish Street, London, W.1, is described in a booklet obtainable from the above address. (2011)

FACTORS AFFECTING THE ECONOMIC OPERATION OF COMMERCIAL VEHICLES*

By G. Waring, A.M.I.Mech.E.† and F. W. Margetts, A.M.I.Mech.E.‡

ALTHOUGH many aspects of design, maintenance and operation were considered, much of the paper dealt with the economics of engine performance. In the authors' opinion this unit provided the most fruitful source of investigation, because whereas the overall efficiency of a good mechanical transmission was as high as 93 per cent, leaving little room for improvement, the thermal efficiency of a good engine of the direct-injection diesel type was stated to be 37 per cent, and for some petrol engines even as low as 23 per cent. Much, however, had been done to secure engine economy by extending the overhaul periods of a commercial vehicle engine which to-day would operate successfully for mileages up to 200,000 between overhauls, thus setting a target for the remainder of the vehicle design.

Dealing first with the size of engine for the work and performance required of it, it was stated that for a heavy goods vehicle 5 or 6 h.p. per ton would be acceptable, whilst for a bus 9 or 10 h.p. per ton would be a reasonable figure in Britain. A private car required something like 25 to 35 h.p. and a 500 cm³ motor cycle utilized 40 to 50 h.p. per ton to give a fairly high degree of road performance. Using these figures as a basis, a 44-seater single-deck bus would require 30 h.p. for the passengers and 75 h.p. for moving the unladen vehicle. Therefore, every effort should be made to reduce the unladen weight because, for every ton saved in weight, the engine power could be reduced by 10. This saving in power, translated roughly, meant a saving of 1 m.p.g.

Concerning the maintenance of chassis units, much had already been done by manufacturers to extend semi-overhaul or intermediate docking periods. For example, the use of rubber bushings in shackle pins provided a probable mileage of 250,000 before replacement. Hard chrome-plated king pins and sleeves employing automatic lubrication would cover at least 250,000 miles without attention, and it was then only necessary to rotate the components through 90 deg in order to duplicate this mileage. Stress was placed on the desirability of frequent examination of lubrication records for all major chassis units at an interval of at least three days to ensure that a unit was not consuming an undue amount of lubricant. Such records, if properly maintained, would indicate immediately leaking glands, worn bearings, and leaking joints.

Extensive tests carried out to determine the relative effects on m.p.g. of the number of bus stops per mile, showed that the effect of 7, 6, or 5 stops a mile gave a proportionate reduction in m.p.g. of 7 per cent. It was suggested that bus stops on city service routes should be positioned on the departure side of traffic light controls at inter-sections, thus avoiding stops whenever the traffic lights were in favour of an approaching vehicle.

Choice of gear ratios was also of vital importance. It was desirable that a bus should re-start easily in second gear when half laden on slight reverse gradients. This was important because, as drivers would not generally use first gear under these conditions, the clutch and gearbox were abused. The use of a close-ratio box to provide easy gear change usually demanded an abnormally low axle ratio to give an easy start on extreme gradients, and this had an adverse effect on fuel consumption.

Concerning the alternative forms of transmission, it was held that, with the present cost of fuel, there was no doubt that the clutch would still lead the field from an economic view. However satisfactory the fluid flywheel might be from a maintenance point of view, all known forms still provided slip and resulted in an increase of 9 or 10 per cent in fuel consumption as compared with a clutch transmission.

Other possible means of reducing costs were suggested. (a) As the 35-gal fuel tank usually fitted on modern vehicles was seldom emptied below the one-third mark with daily refuelling, a 10-gal reduction in capacity was suggested. This would save about 100 lb of dead load. (b) The use of single-deck buses in preference to double-deckers on services with an overall route time of 30 minutes or less, especially on circular routes. Not only was the unladen weight of the vehicle less, but there was less chance of uncollected fares. (c) Decreasing the weight of bus bodies by reducing seat thickness and frame design, the use of lighter weight glass, and a revision in thickness or changes of section for pillars and panels. A scrutiny showed that a saving of 600 lb was possible on these items.

* Abstracted from a paper presented to the Institution of Mechanical Engineers (North West Division).

† Research Engineer, Leyland Motors Ltd.

‡ Production Manager, Research Division, Leyland Motors Ltd.

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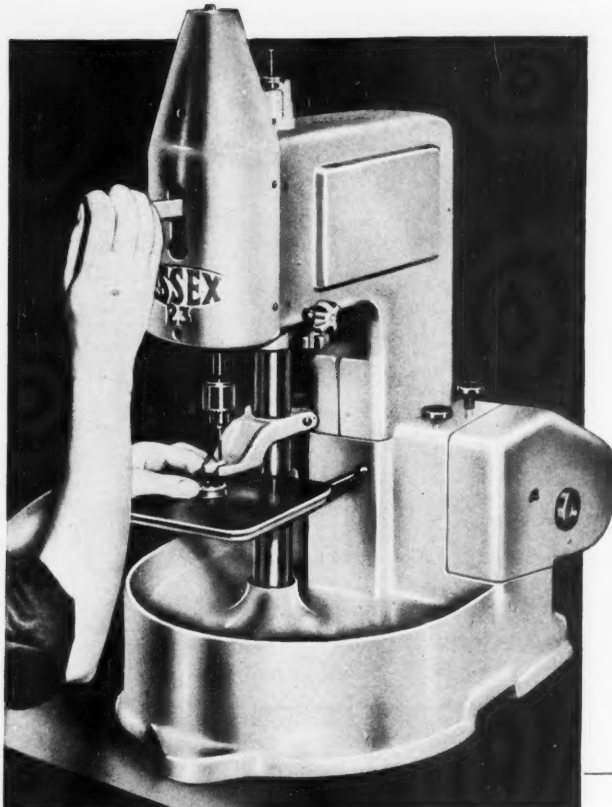
As reported in *Modern Machine Shop*, December, 1951, the Ryan Co. found that when garnet sand crystals break down the resulting smaller crystals still retain sharp cutting edges but that the silica sand disintegrated into relatively smooth crystals having no further value as a cutting agent. Experience has shown also that garnet sand is much less likely to clog machines in damp weather. (2007)



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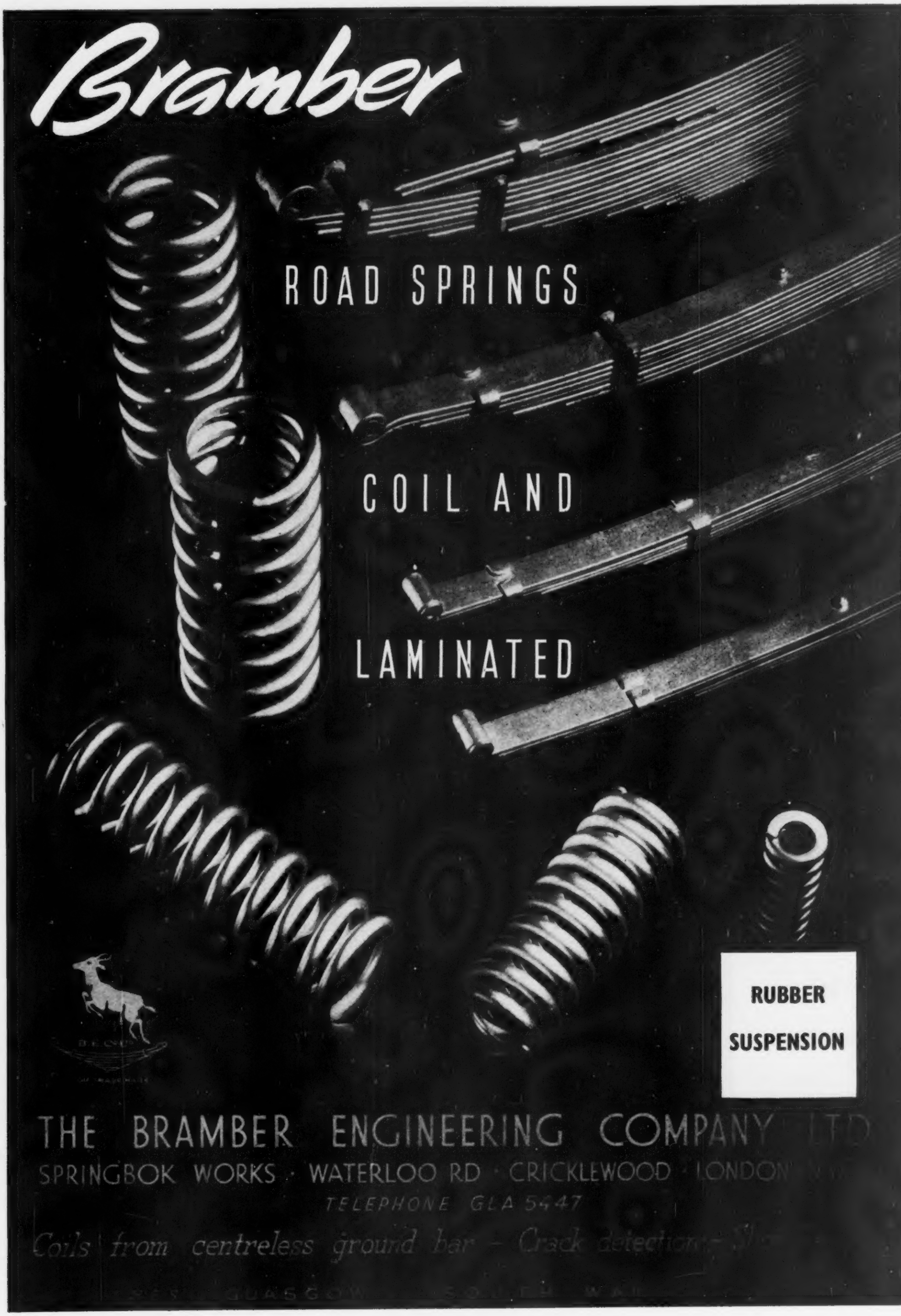
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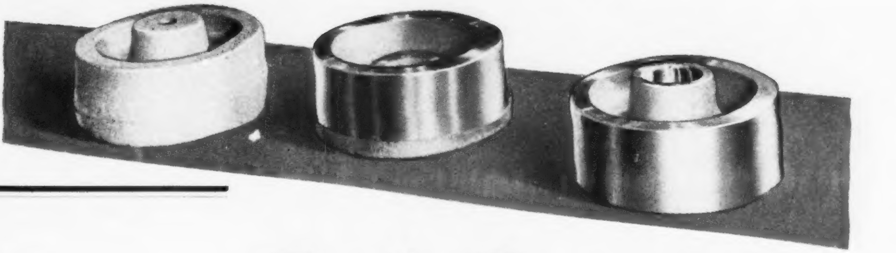

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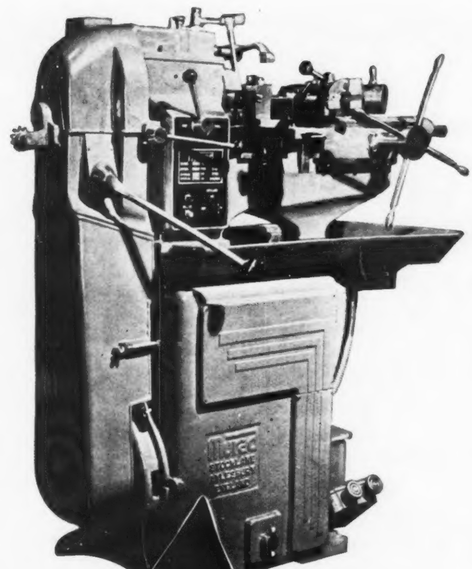

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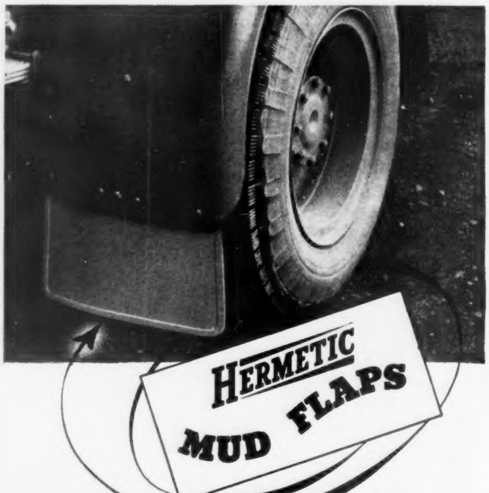

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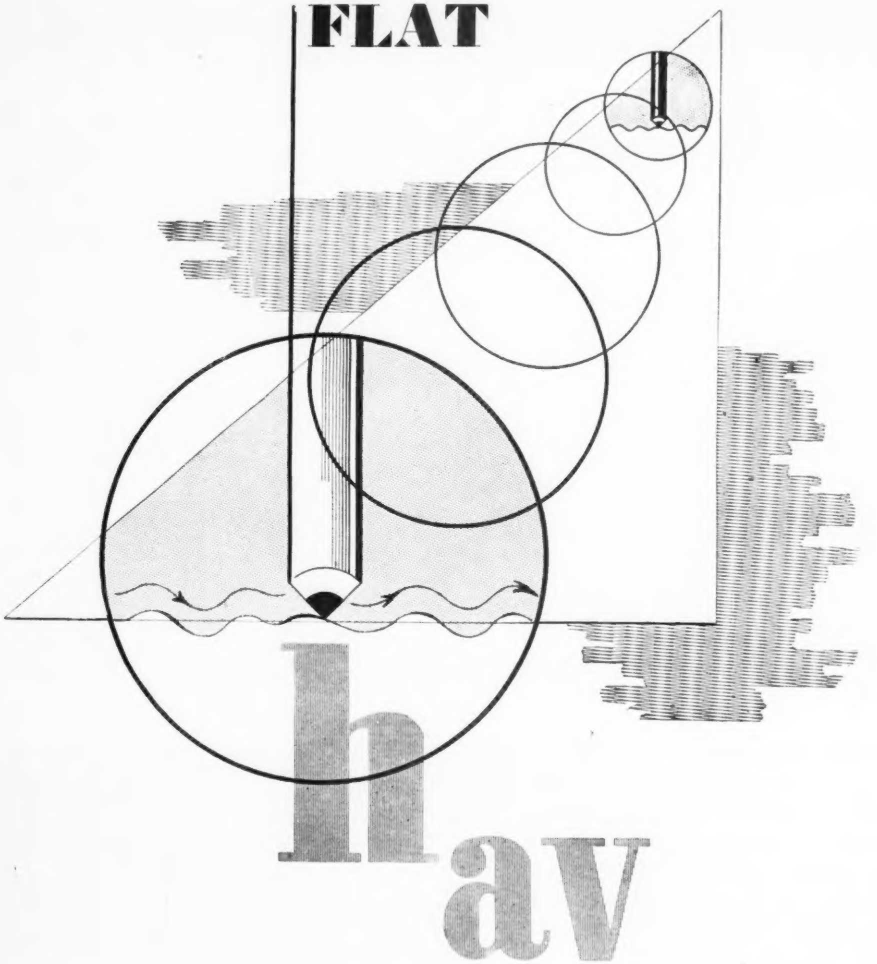

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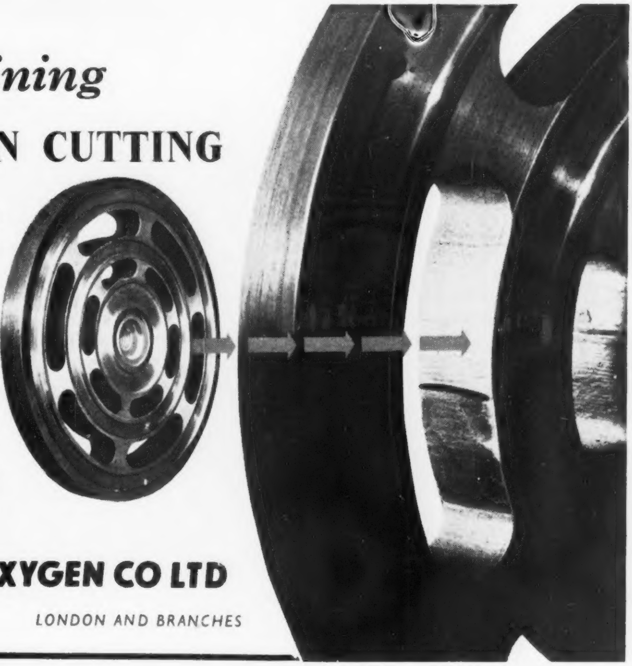

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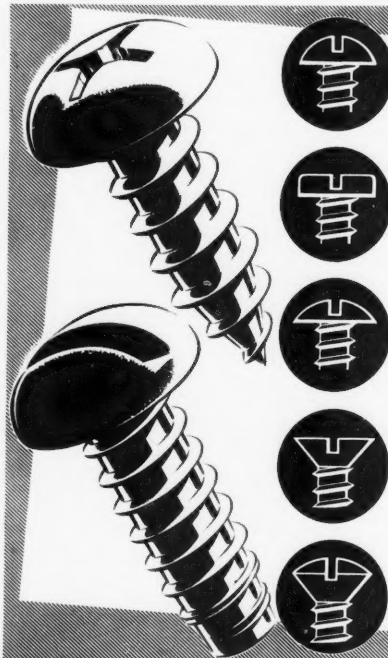

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SITUATIONS VACANT

The engagement of persons answering these advertisements must be made through the local office of the Ministry of Labour and National Service etc. if the applicant is a man aged 18-64 or a woman aged 18-59 inclusive, unless he or she or the employer is excepted from the provisions of The Notification of Vacancies Order, 1952

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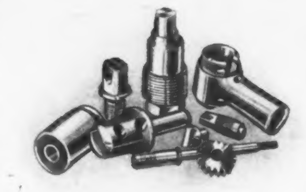

SITUATIONS VACANT

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THE proprietors of British Patent No. 520,192 are prepared to sell the patent or to licence British manufacturers to work thereunder. It relates to piston ring construction.

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